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Latham, New York

MTI-65TR6

EXPERIMENTAL INVESTIGATION OF A
STEAM LUBRICATED JOURNAL BEARING

(Summary Report)

by

J. S. Meacher

Contract Nonr-3731(CO) (FBM)

February 1, 1965

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Contract No. Nonr-3731(00)(FBM)

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ABSTRACT

An externally pressurized, orifice-compensated journal bearing was tested using both nitrogen and steam as lubricant gas. Non-rotating tests were performed to determine load-deflection characteristics, stiffness and flow requirement. Performance with superheated steam, and shaft and bearing temperatures equal to that of the inlet steam, was practically identical to the performance with nitrogen.

Instability similar to pneumatic hammer was observed when shaft temperature was more than 10 F below the inlet steam saturation temperature, using either saturated or wet steam. A minor instability of a different nature occurred when test conditions favored the condensation or accumulation of water droplets in the bearing supply manifold. The bearing was completely unstable when shaft temperatures were more than 25 F below the inlet saturation temperature.

The use of wet steam, saturated steam, or slightly superheated steam, while maintaining shaft temperature 10 F below, at, or 10 F above the inlet saturation temperature resulted in negligible differences in stiffness and load-deflection curves. Some evidence was observed that increased rates of evaporation within the bearing clearance produced slight increases in bearing stiffness.

INTRODUCTION

The use of steam as a bearing lubricant offers a potential increase in reliability and simplicity for steam driven rotating equipment. This potential lies in the elimination of the oil supply sump, pressure pump, filters and cooling systems required by the conventional oil bearing. In addition, the location of the oil lubricated bearing within the equipment is restricted by:

1. The necessity of maintaining oil temperature below 300 to 500 F.
2. The desirability of excluding oil from the working fluid circuit, to prevent fouling of condenser tubes with oil.

The location of steam lubricated bearings, which would not be subject to the above restrictions, could be chosen to result in a simpler and more compact equipment design.

The simultaneous need for fairly high load capacity and the availability of high pressures in steam driven equipment led to the consideration of externally pressurized bearings, in preference to the self-acting type. A previous investigation (Refs. 1 and 2) revealed that condensation within the clearance of an externally pressurized steam lubricated thrust bearing resulted in decreased load capacity, instability, and collapse of the bearing film. These findings indicated that any further study should be devoted to the following operational regimes:

1. Greatly reduced rates of condensation within the bearing clearance
2. Evaporation within the clearance
- 3 The lubricant enters and passes through the bearing as a superheated vapor.

The exploratory investigation reported herein was primarily directed toward determining the engineering feasibility of a steam lubricated journal bearing in the operational regimes stated above. An interim progress report (Ref. 3) was issued in which the test apparatus was described and preliminary test data using nitrogen gas lubricant were reported.

SCOPE

The independent variables which could be manipulated in the application of an externally pressurized, steam lubricated journal bearing, to meet design requirements, are as follows:

1. Supply steam condition - pressure, temperature, quality or superheat, pressure ratio across the bearing
2. Bearing configuration - diameter, length, clearance, nature of flow restricting compensating devices. For orifice compensation, the size, number, configuration, and arrangement of the orifices or feed holes are possible design variables.
3. Thermal environment - shaft temperature, bearing temperature, heat flow into or out of the bearing area.

The dependent performance variables by which the success of the application might be evaluated are:

1. Load capacity - required to support the rotor of the machine being designed, with adequate margin for off-design operation, and acceleration and impact loading.
2. Bearing stiffness - required to permit rotation of the rotor mass at the maximum design speed with no dynamic instability
3. Steam flow demand - desired to be a minimum, since the bearing steam consumption must be charged against the machine when determining overall thermal efficiency.
4. Flexibility - sensitivity to off-design conditions, reaction to transient conditions, ease of start-up and shut-down cycles, static stability.

The formulation of a complete set of design criteria for the steam lubricated journal bearing will require that the effect of each of the independent design variables on the dependent performance variables be determined. This task is beyond the scope of the present work; however, the above considerations have served as a guide for this preliminary investigation. In addition, many of the required relationships have been investigated for gas lubricated bearings. The demonstration of similarity in performance between steam and gas lubricated bearings would constitute a firm foundation for the application of existing gas bearing technology to steam bearing design.

The scope of the present work is outlined below by stating the target range of variables allowed by the test equipment and procedure.

1. Supply steam condition

pressure - 100 psig maximum at steam generator (Pressure loss through the bearing supply system resulted in 70 psig maximum at the bearing.)

temperature - 290 to 450 F

quality - estimated 95 percent to a condition of 160 degrees superheat at the bearing inlet.

pressure ratio - The bearing discharged to atmosphere; pressure ratios of 4, 5, 6 and 7 were attempted.

2. Bearing configuration - Testing has been confined to a single bearing, shown schematically in Fig. 1. The specifications are as follows:

Two rings of 12 feed holes, installed at axial locations .75 inch inboard from each end (The double row bearing provided greater resistance to cocking than would a single row bearing.)

Restrictor orifices installed in removable capsules to allow variable configuration. Either inherent or orifice compensation is possible. Tests were conducted using orifice compensation with recesses or pockets .062 inch diameter, .065 inch deep. Orifice diameters of .014 inch and .025 inch were tested.

3. Thermal environment - The test bearing and shaft were mounted in an insulated enclosure with radiant heaters of 3000 watt total capacity. The hollow shaft contained a 1000 watt resistance heater. Tests were conducted with the shaft temperature above, equal to, and below the inlet steam temperature. The bearing itself formed the inner wall of an inlet steam manifold. As such, it was thermally linked to the inlet steam, and no significant difference between bearing and inlet steam temperatures could be achieved. No provision was made for cooling the shaft or bearing, beyond normal radiation and convection losses.

The bearing was also tested using nitrogen gas as a lubricant, at temperatures of 70 and 400 F. This established a reference performance of an identical gas bearing with which to compare the performance of the steam bearing.

Data were taken to determine the load capacity, stiffness, flow demand, shape of load-deflection curve, and static stability of the bearing. All data were taken without shaft rotation. Preliminary measurements of the pressure within the bearing clearance were made.

Materials were chosen for the test shaft and bearing which were believed representative of those which might be used in practical applications. Thus, some knowledge was acquired of material behavior under prolonged exposure to the steam bearing environment.

RESULTS

Summary

Using nitrogen gas lubricant, the bearing was subject to pneumatic instability with restrictor orifices of .014 inch diameter. The bearing was stable with orifices of .025 inch diameter. Using steam lubricant and .025 inch diameter orifices, the bearing was stable when steam and metal temperatures were 10 degrees above the saturation temperature. Severe instability similar to pneumatic hammer occurred when wet steam was used and metal temperatures were more than 10 degrees below the inlet saturation temperature. A much less severe instability of a different nature occurred at intermediate conditions.

Load-deflection curves using superheated and wet steam were similar to those obtained using nitrogen lubricant. Stiffness and flow demand data obtained during steam tests were in agreement with the predictions of gas bearing theory to a degree at least as high as that of nitrogen test data.

Tests with Nitrogen Lubricant

Test results obtained using nitrogen gas lubricant were previously reported in Reference 3. They are repeated herein for convenient comparison.

The bearing was initially assembled with restrictor orifices of .014 inch diameter. In this configuration the bearing was subject to pneumatic hammer instability when operated with nitrogen gas lubricant at 70 F. A minor re-work was performed to ensure that the feed hole recesses did not contain excessive volume, and larger orifices of .025 inch diameter were installed. Subsequent testing with nitrogen gas at temperatures of 70 and 400 F proved the bearing to be completely stable.

Load deflection data taken with nitrogen at 75 F at pressure ratios of 4, 5, 6, and 7 are shown in Fig. 2. The relative agreement between measured and theoretical predictions of bearing stiffness and flow demand is shown in Fig. 3. The theoretical predictions are based on the analysis of Reference 4. The measured stiffness ranged from nine percent below to 17 percent above the theoretical value. The measured gas flow was from 23 to 29 percent higher than that predicted.

Load deflection data taken with nitrogen at 400 F are shown in Fig. 4. The comparison between measured and theoretical stiffness and flow are shown in Fig. 5. The measured stiffness was from 11 to 19 percent lower than the predicted value. The measured flow was from 18 to 37 percent higher than the theoretical. (A leak in the gas supply system was detected after this test, which produced a positive error in the follow measurements).

Tests in the Superheat Region

Tests were made using superheated steam as the lubricant at a pressure ratio of four, while maintaining the shaft and bearing temperatures as near to the inlet steam temperature as possible. This resulted in a nearly isothermal condition. Tests of this nature were made at temperatures of 450, 400, 350, and 305 F. The steam supply pressure was approximately 59 psia, with a corresponding saturation temperature of 292 F. Thus, the minimum test temperature of 305 F was within the superheat region by a margin of 13 degrees.

Load-deflection data for all of these tests are shown in Fig. 6. The shape of the load-deflection curves for superheated steam is similar to that of the curves obtained with nitrogen, as was expected. The stiffness and bearing steam flow demand are plotted against temperature in Fig. 7, together with values predicted by gas bearing theory based on steam properties. The measured stiffness ranged from 17 percent below to eight percent above the theoretical value. The actual stiffness appeared to be more sensitive to temperature changes than was predicted by the theory. The measured steam flow was within five percent of the predicted value at all temperatures. The gas bearing theory evidently predicts the stiffness and flow demand of a superheated steam lubricated bearing at least as well as it does those of a nitrogen lubricated bearing.

The bearing was completely stable during these tests. No trace of pneumatic hammer or other instability existed.

Tests Near the Saturation Temperature

Several situations are possible when operating near T_s , the saturation temperature.*

*The saturation temperature referred to is that corresponding to the bearing inlet or supply pressure, measured upstream from the restrictor orifices.

Those that were investigated are enumerated below:

1. Metal (bearing and shaft) considerably below T_s , no heat conduction into the bearing area; supply steam wet.
2. Metal slightly below T_s , with heat flow into the bearing area;
 - a. supply steam wet
 - b. supply steam saturated
 - c. supply steam superheated

Tests were conducted at each of the above conditions at pressure ratios of four and five. (A pressure ratio of six could not be maintained at the bearing due to pressure drop through the supply line and excessive fluctuations of the steam generator pressure.)

Load deflection data were obtained at each test condition, from which bearing stiffness values were determined. The load-deflection curves for conditions 2, 3, 4, and 5 at a pressure ratio of four are shown in Fig. 8, and conditions 6, 7, and 8 are shown in Fig. 9. The load deflection curve obtained with nitrogen at 400 F and a pressure ratio of four is also shown in Fig. 9 for comparison. The general shape of all the curves obtained with steam is not significantly different from that of the nitrogen curve. There was no tendency toward sudden collapse of the film at any of these test conditions.

The load deflection curves for the various conditions at a pressure ratio of five are shown in Figs. 10 and 11. Again, the curve for nitrogen at 400 F and a pressure ratio of five is shown for comparison. The shapes of the various curves for steam do not differ significantly among themselves, nor do they differ greatly from the nitrogen curve. Again, there was no tendency toward collapse of the film. Indeed, the steam bearing appeared to maintain its stiffness to higher eccentricities than did the nitrogen bearing.

A considerable shift of the curves along the deflection axis from one condition to another is evident. This was probably due to both differential thermal expansion and a gradual build-up of mineral deposits on the shaft. The most severe build-up occurred just outboard of the bearing under the deflection sensing probes. The resulting shifts did not, however, affect the validity of stiffness determinations.

Flow measurements were also taken. However, in those cases where wet steam was supplied to the bearing, the indicated flow is not a true measure of the mass flow passing through the flow measuring nozzle. The quality of the steam was not determined, so entrained moisture constituted an additional mass flow fraction not indicated by the measuring device. Nevertheless, the approximate flow measurements allow a comparison of the volume flow demand made by the bearing on the steam supply system at various operating conditions.

Neither bearing supply pressure nor temperature serve as a convenient reference for plotting the data. The results are therefore presented in tabulated form. Table I presents the data for a pressure ratio of four, and Table II presents data for a pressure ratio of five.

Column one of Table IV corresponds to condition 1 described above, viz., metal temperatures considerably below T_g , no heat supply to the bearing area, and wet supply steam. The test apparatus was heated prior to admitting steam to the bearing. Shaft temperature was 265 F and bearing temperature 266 F at this point. After supplying wet steam, the bearing temperature quickly increased to about 280 F. (As stated previously, the bearing proper forms the inner wall of an inlet steam manifold which supplies the 24 restrictor orifices. This is shown schematically in Fig. 1. The bearing temperature therefore tended to follow the inlet steam temperature.) The bearing was completely unstable, oscillating in both random and periodic fashion throughout the entire clearance space. Applying load to the bearing did not reduce the instability, but did tend to change random oscillation to an ordered form, similar to pneumatic hammer. Fig. 12 shows an oscilloscope photo of the bearing "orbit" at this condition, with the clearance circle inscribed for reference.

The pressure within the bearing supply manifold was very slow in increasing to the desired level. There was undoubtedly condensation taking place within the manifold. The resulting moisture, plus that entrained in the incoming wet steam, produced an extremely wet mixture at the restrictor orifices, with the probability of water droplets entering the orifices and clogging them in random fashion. This might explain the random motions of the bearing, but would not explain the ordered oscillations shown in Fig. 7. Similar results were obtained at a pressure ratio of five under the conditions of column one, Table II.

Traces of instability of two different types were noted during some of the tests listed in Tables I and II. Under the conditions of column two (wet steam, metal temperature below T_g), the bearing tended to go into the "pneumatic hammer" type of instability whenever the supply pressure was adjusted. Whenever metal temperatures were more than 10 degrees below T_g and the inlet steam was wet, the bearing was subject to the same type of instability over a wide range of loads. The action of the bearing during these instabilities appeared to be identical to that observed during the initial tests with nitrogen. Pneumatic instability of gas bearings frequently arises from excessive storage volumes in or linked to the clearance space. It can be surmized that condensation within the clearance of a steam bearing could act as a "storage volume", producing a similar effect.

The second type of "instability" consisted of momentary excursions of the bearing from its equilibrium position. These excursions occurred at random intervals, in the direction of the applied load, with a maximum amplitude of about .0005 inch, equivalent to an eccentricity ratio of 0.2. An audible hiss frequently accompanied these excursions. Momentary decreases of the pressure in the clearance space (downstream from a restrictor orifice) also accompanied the excursions. This instability was most pronounced when supplying wet steam to the bearing, and when the radiant heaters surrounding the bearing were not operating. The amplitude and frequency of occurrence decreased as the inlet steam condition moved toward the superheat region, and as the radiant heating around the bearing was increased. Shaft heating appeared to have little effect on the instability. These circumstances indicate that the instability is excited by conditions allowing the accumulation or condensation of water within the steam supply manifold surrounding the bearing. The random passage of water droplets through individual orifices would cause momentary blockage and consequent reduction of the local downstream pressure within the clearance space. This could cause a momentary shift of the bearing.

The stiffness data of Table I indicate that the various conditions imposed on the bearing had negligible effects on the bearing stiffness at a pressure ratio of four. The average stiffness for the seven conditions where stiffness was measured is 116,700 lb/inch. This was within nine percent of the theoretical value (129,000 lb/inch) at 290 F for dry steam properties. All seven measured stiffness values are within 3-1/2 percent of the average. This deviation is

certainly within the range of experimental error. However, note that one of the highest values (120,600 in column eight) resulted from admitting wet steam to a bearing heated to a temperature higher than the saturation temperature. This condition would undoubtedly result in evaporation within the clearance space. If the evaporation takes place in a manner to augment the circumferential pressure gradient within the clearance of a loaded bearing, then it should produce a stiffer bearing. However, the low stiffness from condition 7, where evaporation should also have occurred, does not substantiate this hypothesis. Another factor which must be considered is the change in clearance due to the difference between shaft and bearing temperatures. For condition 8, the temperature difference is sufficient to cause about a seven percent decrease in clearance. This could well account for the increased stiffness observed at this condition.

At a pressure ratio of five, larger differences were noted between the stiffnesses at the various conditions. The average stiffness for the eight values shown in Table II is 182,300 lb/inch. This is within 13 percent of the theoretical prediction (162,000 lb/inch) for dry steam at 306 F, using gas bearing theory. The highest value (217,000 for condition 7) was 19 percent above the average. This is also a condition where evaporation probably occurred in the film. The lowest value (165,000 for condition 2) was nine percent below the average and resulted from a condition where wet steam was admitted to a bearing with metal temperatures below the inlet saturation temperature. Again, some of the variation can be attributed to experimental error.

It is probable that evaporation occurred somewhere within the clearance space during all tests with wet steam. The minimum shaft temperature recorded during tests at a pressure ratio of four was 281 F, which corresponds to a saturation pressure of about 50 psia. Thus, at all points in the film where the pressure was below 50 psia, entrained moisture would have been at a lower temperature than the shaft and heat transfer would have occurred to produce evaporation. Similarly, a minimum shaft temperature of 291 F was recorded during tests at a pressure ratio of five. The corresponding saturation pressure was 59 psia.

The flow indications recorded in Table I do not vary significantly except for those cases where wet steam was used and higher heating was applied. The conversion of entrained moisture to vapor, either within the bearing supply manifold or in the clearance space, evidently contributed to satisfying the flow

demand of the bearing. The entrained moisture, as previously noted, was not indicated by the flow measuring nozzle. Similar results were obtained at a pressure ratio of five.

Miscellaneous

On one occasion, steam was admitted to the bearing with the entire test apparatus near room temperature. Severe vibrations and instability of the bearing resulted. These were of sufficient strength to excite vibration of the entire test enclosure. Adding superheat to the inlet steam did not reduce the instability. External heating gradually reduced the instability. The condition appeared similar to that cited previously, where metal temperature 25 degrees below saturation temperature caused instability of lesser magnitude.

No significant degradation of the bearing and shaft materials was apparent during the course of the investigation. Mineral deposits and stains occurred on both the shaft and bearing after six to eight hour periods of operation. The mineral deposits were removed with a quick rinse in dilute sulfuric acid followed by a thorough washing in water. Some permanent staining did not respond to this treatment. Examination of the shaft after cleaning revealed some slight reduction of surface polish or reflectivity in some areas. It was not determined whether this was due to erosion or staining. Nor could any erosion effects be discerned on the restrictor orifices.

Pressure measurements were made in the clearance space to define the axial pressure profile at a pressure ratio of five with dry steam. However, the fluctuation in steam generator pressure could not be controlled during the time required to make the traverse. The resulting variations in supply pressure produced distorted pressure profiles.

CONCLUSIONS

Operation in the Superheat Region

An orifice compensated, externally pressurized journal bearing can be successfully operated using superheated steam as lubricant with bearing and shaft temperatures equal to the inlet steam temperature. The load-deflection characteristics, stiffness near zero eccentricity, and lubricant flow demand can be predicted by gas bearing theory as accurately as can similar properties of a gas lubricated bearing. No peculiarities in performance occur so long as the inlet steam and metal temperatures are more than 10 degrees above the inlet saturation temperature.

The maintenance of the conditions specified above would require careful consideration of the heat conduction paths around the bearing area in practical applications. Excessive heat conduction away from the bearing area through the shaft would reduce the journal temperature.

Operation Near the Saturation Temperature

An orifice compensated bearing, of a configuration indicated by Fig. 1, will be subject to instability similar to pneumatic hammer when operated with wet steam and metal temperatures more than 10 degrees below the inlet saturation temperature. Metal temperatures 25 degrees or more below the inlet saturation temperature will apparently result in complete loss of load carrying ability and severe vibrations. The performance observed under these conditions during this investigation could be attributed in part to condensation within the bearing supply manifold. As such, it would not be considered a property of the bearing orifice-clearance system itself. However, the configuration of the test bearing seems representative of what might be desired for practical application. The test results reported herein therefore seem pertinent to practical applications of steam bearings under similar test conditions.

When metal temperatures are no less than 10 degrees below nor more than 20 degrees above the inlet saturation temperature, the bearing stiffness, flow demand, and load deflection characteristics at moderate pressure ratios will not differ significantly from those predicted by gas bearing theory. This is true for operation with wet, dry, or slightly superheated supply steam.

There is some evidence that the bearing stiffness increases slightly as increasing degrees of evaporation occur within the clearance. This requires heat flow into the bearing area. The usual bearing configuration (with a steam supply manifold surrounding the bearing proper) would require that this heat entered through the shaft. The consequent temperature difference between shaft and bearing must be considered. The concept of evaporation within the bearing clearance space necessitates higher metal temperatures. A simultaneous requirement for equal bearing and shaft temperatures means that the bearing temperature must be independent of the inlet steam temperature. This will require close attention to the layout and design of the steam supply system for the several orifice locations in the bearing.

Bearing design and operating conditions which would allow droplets of water to enter individual restrictor orifices will result in random excursions of the shaft from equilibrium position. The magnitude of these excursions would probably depend on the number of orifices around the bearing. Probably larger excursions would result in a bearing with fewer orifices. Excursions equivalent to eccentricity ratios up to 0.2 have been observed with 12 orifices per row.

"Start-up" of steam bearings from the cold condition is a potential problem area. If metal temperatures are brought up to the operating level prior to admitting steam to the bearing, no difficulties will be encountered. However, sudden admission of steam with cold bearing and shaft bodies could result in severe vibration over extended periods — until metal temperatures increase to the normal operating level. This will necessitate additional study of the thermal environment surrounding steam bearings in practical applications.

Stellite 6 appears to be a satisfactory material for use in steam bearing applications. However, only operation at moderate pressure ratios has been investigated. Higher pressure ratios, with consequent higher flow velocities, may yet result in erosion. The flame-sprayed bronze coating for the bearing is also satisfactory, when the configuration of feed holes does not result in high velocities immediately adjacent to the bronze surface. This material is also limited to moderate temperatures. Operation at temperatures much above 500 F would be marginal for this material.

RECOMMENDATIONS

Further investigation in the superheat region at moderate pressure ratios does not appear to be necessary. Sufficient work should be done, however, to demonstrate stable and predictable operation at higher pressure ratios, on the order of eight to ten. This would provide a firm foundation for the design of bearings with higher load capacity. Extending the work to still higher pressure ratios does not seem warranted, for two reasons:

1. The correlation between gas bearing performance and gas bearing theory deteriorates at higher pressure ratios; consequently, predicting the performance of steam bearings will be subject to the same degree of uncertainty.
2. Erosion problems would probably become significant.

A more exact definition of the minimum shaft and bearing temperatures compatible with stable operation near the saturation temperature is desirable. This entails the following:

1. Positive verification that the momentary excursions observed during these tests are due to momentary clogging of individual orifices by water droplets washed from the interior surface of the supply manifold. If so, effect design changes in the bearing to prevent such occurrence. If not, isolate the cause.
2. Determine whether the "pneumatic hammer" type of instability observed during wet steam tests can be eliminated or reduced by modification of the feeder hole recesses and restrictor orifices.
3. Determine means of making the temperature of the bearing itself independent of the inlet steam temperature.
4. Modify the steam generator control system to allow regulation of pressure within ± 1 psi or better. The existing steam generator control system produces fluctuations of up to ± 5 psi. In the pressure range being investigated, a change of 1 psi produces a corresponding change of about 1 F in saturation temperature.
5. Determine whether or not the same temperature margins are applicable at pressure ratios up to 10. (This would necessitate a new steam generator. The maximum working pressure of the existing unit is 100 psig. The remainder of the test apparatus could be used unchanged.)

If the bearing temperature can be made independent of the inlet steam temperature, a more exact determination of the effects of evaporation within the clearance space should be made. A modification to the bearing supply manifold system will be required to accomplish this.

The improvement of the steam generator pressure control system would also allow meaningful measurements of pressure within the clearance space. This would allow better determination of the degree of evaporation occurring in the clearance.

Sufficient foundation now exists for the design of a rotating shaft-bearing system. The validation of gas bearing rotating stability criteria in steam bearings is the next requirement in working toward practical applications. A background of high speed rotating operation over a limited range of pressure and temperature should be built up. A simple two-bearing test rig, without provisions for external loading, should be adequate for this purpose.

At some point, the effects of reduced ambient pressure should be determined. This condition would be encountered by a bearing installed in the exhaust chamber of a condensing turbine.

DISCUSSION OF TEST APPARATUS AND METHODS

Test Apparatus

The general nature of the test bearing is shown schematically in Fig. 1. The inner shell, or bearing proper, was machined from SAE 4140 steel. The bore surface was flame sprayed with phosphor bronze (Metco Spraybronze "P") prior to final lapping. Twenty-four orifice locations were installed. The restrictor orifices were mounted in removable capsules. The configuration used during the tests described herein is shown in Fig. 13. An outer shell was mounted over the bearing itself, to form a steam supply manifold for the several orifices. Silicone rubber "o"ring seals were installed at each end of the manifold. Both the manifold and the bearing (excluding the bore surface) were electroless nickel plated to prevent corrosion. Fig. 14 is a photograph of the bearing assembly.

The clearance and orifice size used during these tests were such as to yield a dimensionless feeding parameter, λ_s , very near the optimum value for pressure ratios from four to seven, and temperatures from 300 to 400 F, using steam lubricant. For these conditions, $.84 < \lambda_s < 1.38$; the optimum value of λ_s is 1.5.

The hollow test shaft was machined from 4140 steel, hardened to 34 Rc, and plasmarc coated with Stellite 6 on the journal surface prior to finish grinding and lapping.

A 1000 watt sheathed wire resistance heater was coiled inside the shaft, in intimate contact with the bore surface. The power to this shaft heater was regulated with a 1 kw Variac. Also installed in the shaft were two thermocouples, and a pressure tap and pressure transducer for sensing pressure in the clearance space. By translating and rotating the shaft within the bearing, axial and circumferential pressure profiles could be determined. The shaft was rotated in support bearings mounted on stub sections bolted to each end of the journal section. Light series, 40 mm ball bearings were first used for support bearings since rotating tests were also being considered. Excessive growth after prolonged exposure to high temperature rendered these support bearings unacceptable. Hardened steel bushings with Electrofilm coatings on inner and outer diameters were substituted. Fig. 15 is a photograph of the shaft assembly prior to testing, with the ball bearings installed. The interior details of the shaft are shown in Fig. 16, which is an assembly drawing of the entire test apparatus.

The shaft support bearings, in turn, were mounted in sleeves welded to the walls of the test enclosure. Axial traverses were accomplished by sliding both the shaft and the support bearings in the sleeves by means of jack screws. The enclosure was essentially a box of welded steel plate construction with a channel iron foundation. The entire structure was electroless nickel plated to prevent corrosion. The enclosure and bearing were heated by means of six quartz-rod infrared heaters installed inside the enclosure. Lava plugs inserted in holes in the enclosure walls supported the ends of the heaters. Fig. 17 is a photograph showing the enclosure and arrangement of heater rods. The heaters were rated at 500 watts each at 115 volts. The six rods were wired in parallel to a three kilowatt Variac, which allowed excellent adjustment of the heat input.

Load was applied to the bearing by a pneumatic cylinder mounted underneath the base of the enclosure, as shown in Fig. 16. Cylinder pressure was supplied by high pressure nitrogen through a pressure reducer and Grove pressure regulator. The cylinder exerted a downward pull on the bearing through an articulated rod connected to a lug on the bearing outer shell. Teflon lined Uni-bal ends were used on the rod to minimize any tendency of the loading system to cock the test bearing. When supported on a lubricant film, the bearing was free to slide along the shaft to a point where the load line was perpendicular to the shaft.

Steam was supplied by an 18 kw electric steam generator rated at 60 pounds per hour capacity at 100 psig. Pressure regulation was effected by an adjustable pressure control switch which turned heater power on or off. The device appeared to have a minimum "dead band" of about ± 4 to 5 psi. An electric superheater was built from stainless steel tubing. The tubing acted as both steam flow passage and resistance heating element. The unit was rated at approximately 3 kw capacity and input power was regulated by a 3 kw Variac. The unit provided very good regulation of superheat temperature.

Tap water was used in the steam generator. An ion exchange water softener was the only treatment given the feedwater. As mentioned previously, a build-up of mineral deposits on both shaft and bearing was noted after prolonged testing with wet steam. Fig. 18 shows the condition of the shaft after such a test, and Fig. 19 the bearing after the same test. Most of these deposits could be removed by rinsing with dilute sulfuric acid.

The water pressure at the tap was sufficient to introduce feedwater into the steam generator under most test conditions, so a feedwater pump was not used. On those occasions when tap water pressure dropped to about 100 psig, difficulty was encountered in maintaining steam generator pressures above about 90 psig.

The steam line from the superheater to the bearing incorporated a flow measuring nozzle and valving to bypass the steam to a spray-jet type condenser during warm-up periods. The final run into the bearing was installed to impose minimum restraint on the test bearing. The configuration is shown in Fig. 20, which is an overall view of the entire test facility. The desired flexibility was achieved by installing a long length of 1/4-inch tubing in pendulum fashion. This allowed the bearing to move freely, but, in conjunction with the bypass valving, introduced considerable pressure drop. The entire steam supply line was well insulated.

Measurement Systems

Capacitance type distance measuring equipment was used for bearing clearance measurements. The probes were secured to the bearing with clamps shown in Fig. 19. These were mounted at 45 degree locations on either side of the load line, rather than in the usual vertical and horizontal positions. This prevented interference between the probes and the enclosure walls. As a consequence, bearing deflections under load occurred at a 45 degree angle on an "x-y" oscilloscope display of the probe output signals. Typical load-deflection data are shown in Fig. 21. This consists of a multiple exposure photograph of the oscilloscope screen. Each exposure was made with a different pressure applied to the loading cylinder. A contact or "bottomed" point was always taken for reference. "Bearing lift" was determined by measuring the distance, on the photograph, between the reference point and each of the other points and multiplying by the probe calibration factor. The probe-oscilloscope combination was calibrated with a micrometer fixture at temperatures from 70 to 450 F. No discernable change in linearity or calibration factor occurred over this temperature range.

The load-deflection curves were obtained by plotting the data obtained above. A graphical determination of the bearing stiffness was made by measuring the slope of the load-deflection curve near the minimum load point. Any scatter of the load-deflection points made the determination of stiffness by this method somewhat arbitrary and subject to uncertainty.

The steam flow was measured with a nozzle installed downstream from the superheater in accordance with ASME Power Test Code recommendations. The nozzle ΔP was measured with a 0-100 inch water differential pressure gage. The nozzle was calibrated with nitrogen against a precision flowmeter, prior to use with steam. This system did not allow accurate mass flow determinations when measuring the flow of wet steam.

Temperatures were measured with special grade copper-constantan thermocouples. A Honeywell 0-500 F multipoint recorder was used for temperature readout. The thermocouples and recorder were tested with boiling distilled water and found accurate within two degrees or better. The several thermocouple installations are shown in Fig. 22. The nozzle and bearing inlet steam thermocouples were installed in a manner to reduce conduction errors to a minimum. The bearing and bearing-manifold-steam thermocouples, however, were probably subject to some conduction errors. A thermometer was also installed in the superheater discharge fitting.

The bearing supply pressure was measured by a 0-100 psig gage connected to a tap in the supply manifold, indicated in Fig. 22. Upstream pressure at the flow nozzle was measured with a 0-100 psig Barton gage, and a 0-150 psig Heise gage was used to measure the load cylinder pressure. The pressure transducer mounted in the shaft was a CEC model 4-137 strain gage type, and was read out on an Ellis Associates model BAM-1 bridge amplifier meter. The transducer-meter combination was calibrated in place at the operating temperature by traversing the shaft to expose the sensing point and applying a known pressure.

REFERENCES

1. Orcutt, F. K., "Experimental Investigation of Two-Phase Flow in Thrust Bearings", MTI-62TR40, Prepared for ONR under Contract No. Nonr-3731(00)(FBM), Jan, 1963.
2. Orcutt, F. K., "Experimental Investigation of Two-Phase Flow in Thrust Bearings - Part II, Cooled Rotating Surface Bearing", MTI-63TR28, ONR Contract No. Nonr-3731(00)(FBM), July 1963.
3. Meacher, J. S., "Experimental Investigation of a Steam Lubricated Journal Bearing", MTI-64TR40, Contract Nonr-3731(00)(FBM), August, 1964.
4. Lund, J., Wernick, R. J., and Malanoski, S. B., "Analysis of the Hydrostatic Journal and Thrust Bearing for the NASA A6-5 Gyro Gimbal Bearing", MTI-62TR26, Contract NAS 8-2588, Subcontract 213-03-72120-6244, October, 1962.

TABLES

TABLE I
TEST RESULTS NEAR THE SATURATION TEMPERATURE

Pressure Ratio = 4
 Supply Pressure = 58.8 psia
 Saturation Temperature, $T_S = 292^{\circ}\text{F}$

Test Condition	1	2	3	4	5	6	7	8
Nominal Metal Temperature	$T_S - 25$	$T_S - 10$	$T_S - 10$	$T_S - 10$	$T_S - 10$	T_S	T_S	$T_S + 20$
Nominal Steam Condition	wet	wet	dry	$T_S + 10$	$T_S + 30$	dry	wet	wet
Measured Stiffness, lb/in.	--	116,300	120,000	113,600	113,100	118,600	114,600	120,600
Indicated Flow, lb/hr.	--	27.4	27.2	27.2	26.2	24.0	22.8	18.6
Shaft Heater Power, watts	0	86	52	38	.5	105	160	0
Enclosure Heater Power, watts	0	0	0	0	0	380	380	3000
Inlet Steam Temperature, $^{\circ}\text{F}$	285	292	294	301	322	292	292	292
Bearing Temperature, $^{\circ}\text{F}$	280	290	290	290	292	290	290	311
Shaft Temperature, $^{\circ}\text{F}$	265	282	281	281	281	292	294	324
ΔT , Shaft-to-Bearing	-15	-8	-9	-9	-11	+2	+4	+13

TABLE II
TEST RESULTS NEAR THE SATURATION TEMPERATURE

Pressure Ratio = 5
Supply Pressure = 73.5 psia
Saturation Temperature, $T_s = 306^\circ\text{F}$

Test Condition	1	2	3	4	5	6	7	8	9
Nominal Metal Temperature	$T_s - 25$	$T_s - 10$	$T_s - 10$	$T_s - 10$	$T_s - 10$	T_s	T_s	$T_s + 20$	$T_s + 10$
Nominal Steam Condition	wet	wet	dry	$T_s + 10$	$T_s + 30$	dry	wet	wet	dry
Measured Stiffness, lb/in.	--	165,600	182,200	170,600	186,400	182,500	217,200	181,900	172,100
Indicated Flow, lb/hr.	--	36.2	36.4	36.6	35.6	32.7	33	29.6	31
Shaft Heater Power, watts	0	178	46	32	0	47	0	168	52
Enclosure Heater Power, watts	0	0	0	0	0	354	1100	2129	129
Inlet Steam Temperature, $^\circ\text{F}$	306	306	310	318	336	313	306	306	306
Bearing Temperature, $^\circ\text{F}$	302	303	303	304	305	305	305	308	308
Shaft Temperature, $^\circ\text{F}$	278	292	291	291	292	306	308	319	312
ΔT , Shaft-to-Bearing	-24	-11	-12	-13	-13	+1	+3	+11	+4

FIGURES

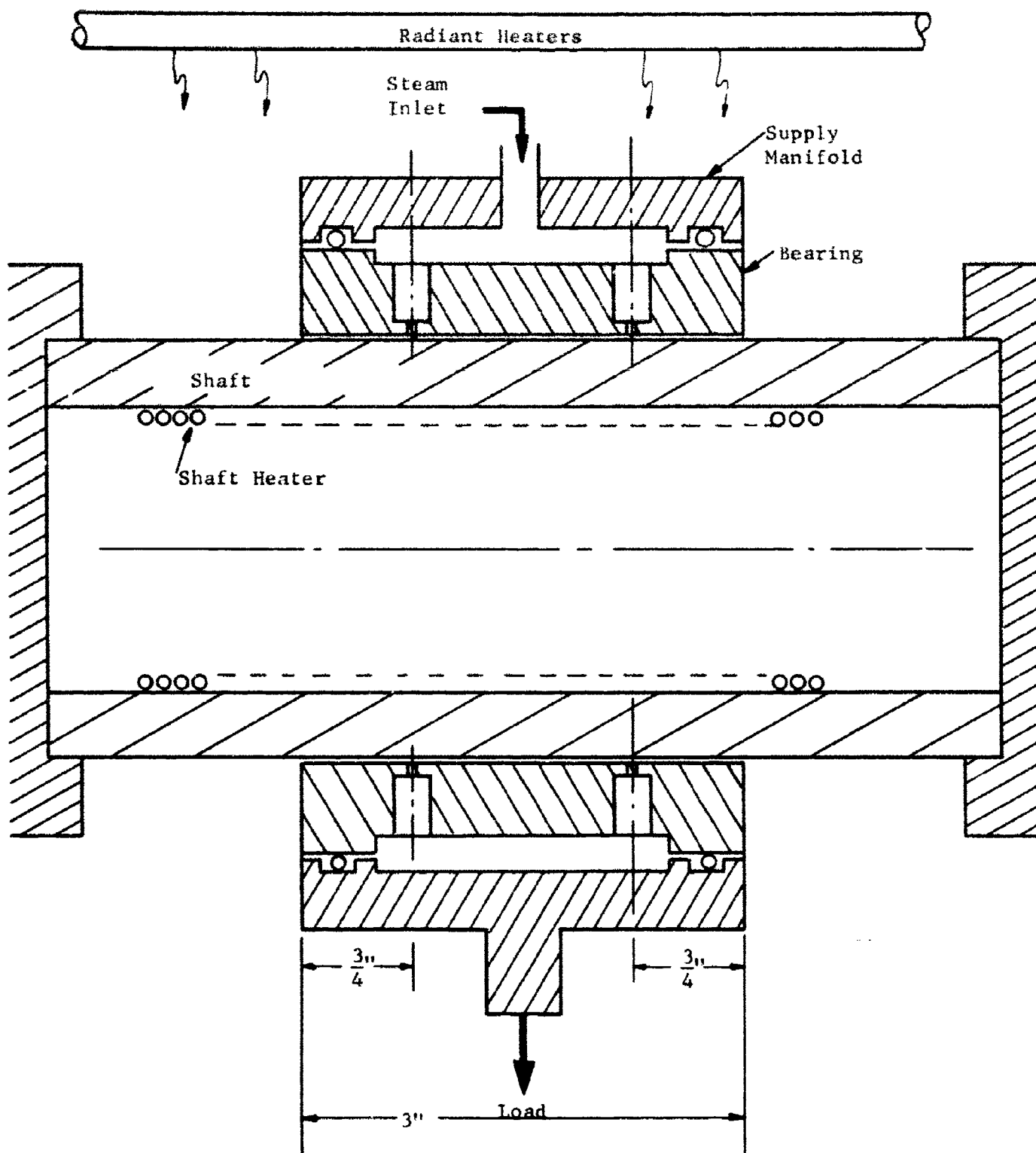


Fig. 1 Schematic of Test Bearing

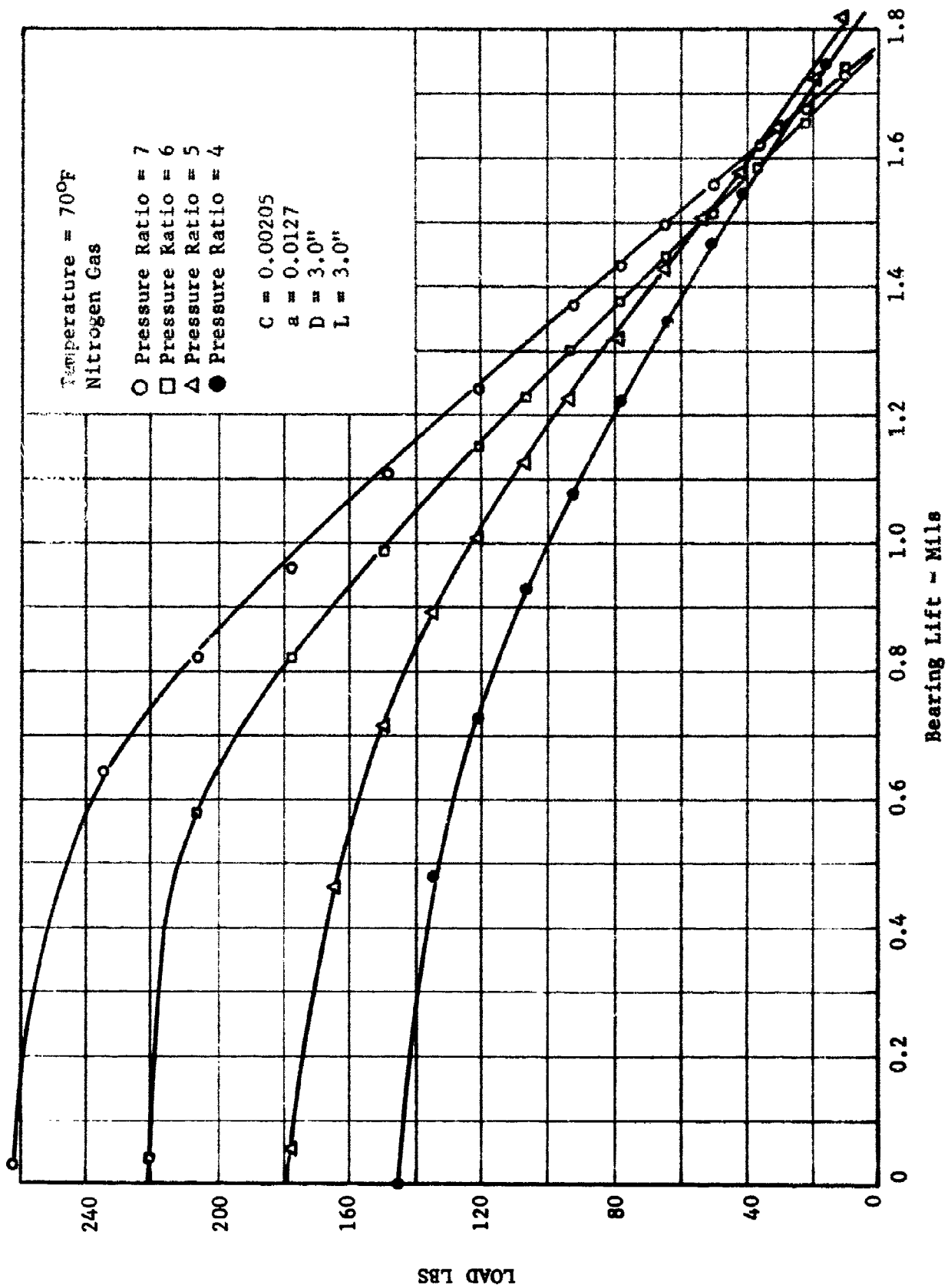


Fig. 2 Load-Deflection Curves, Nitrogen Lubricant, 75°F

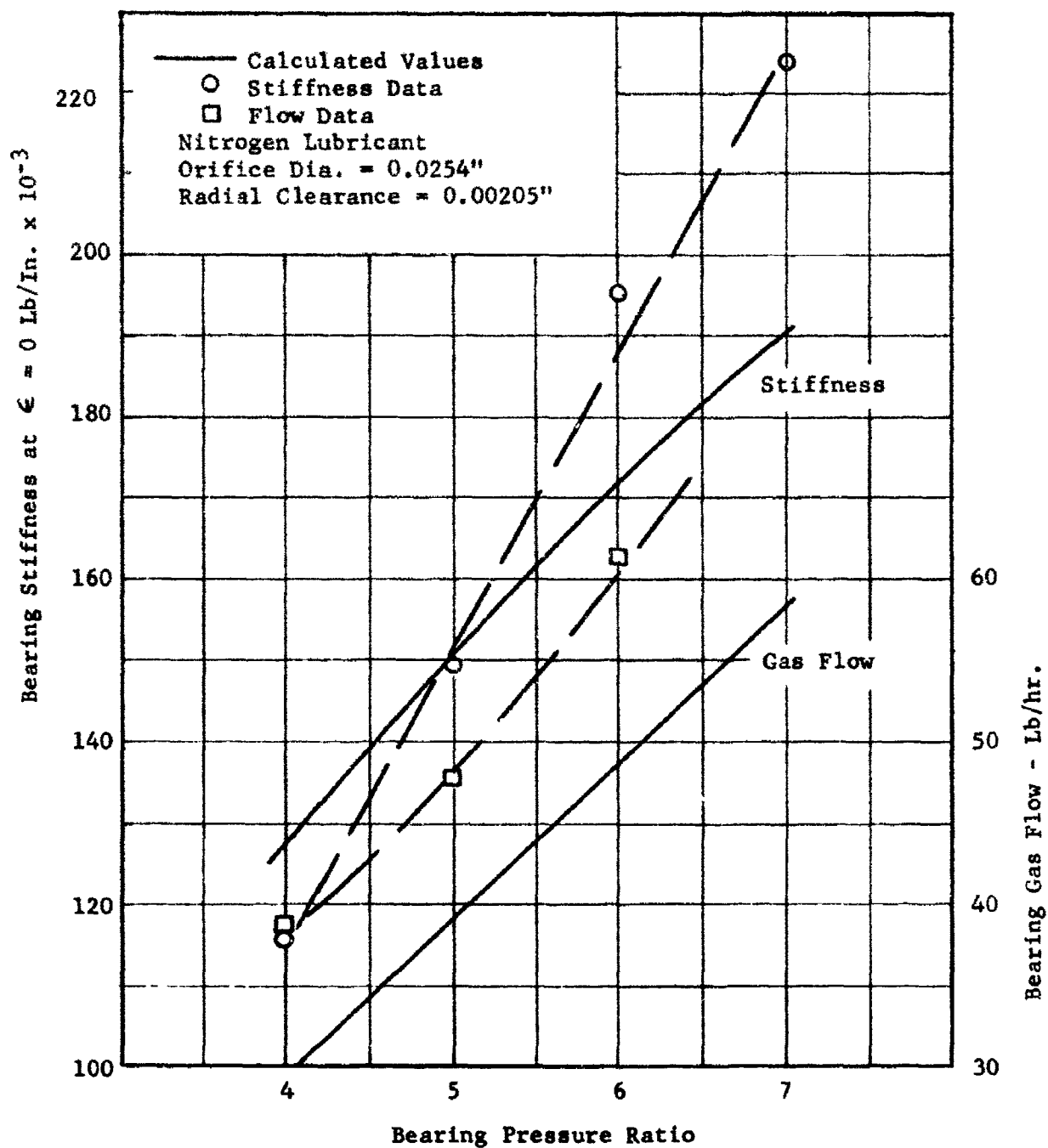


Fig. 3 Stiffness and Gas Flow,
Nitrogen Lubricant, 75°F

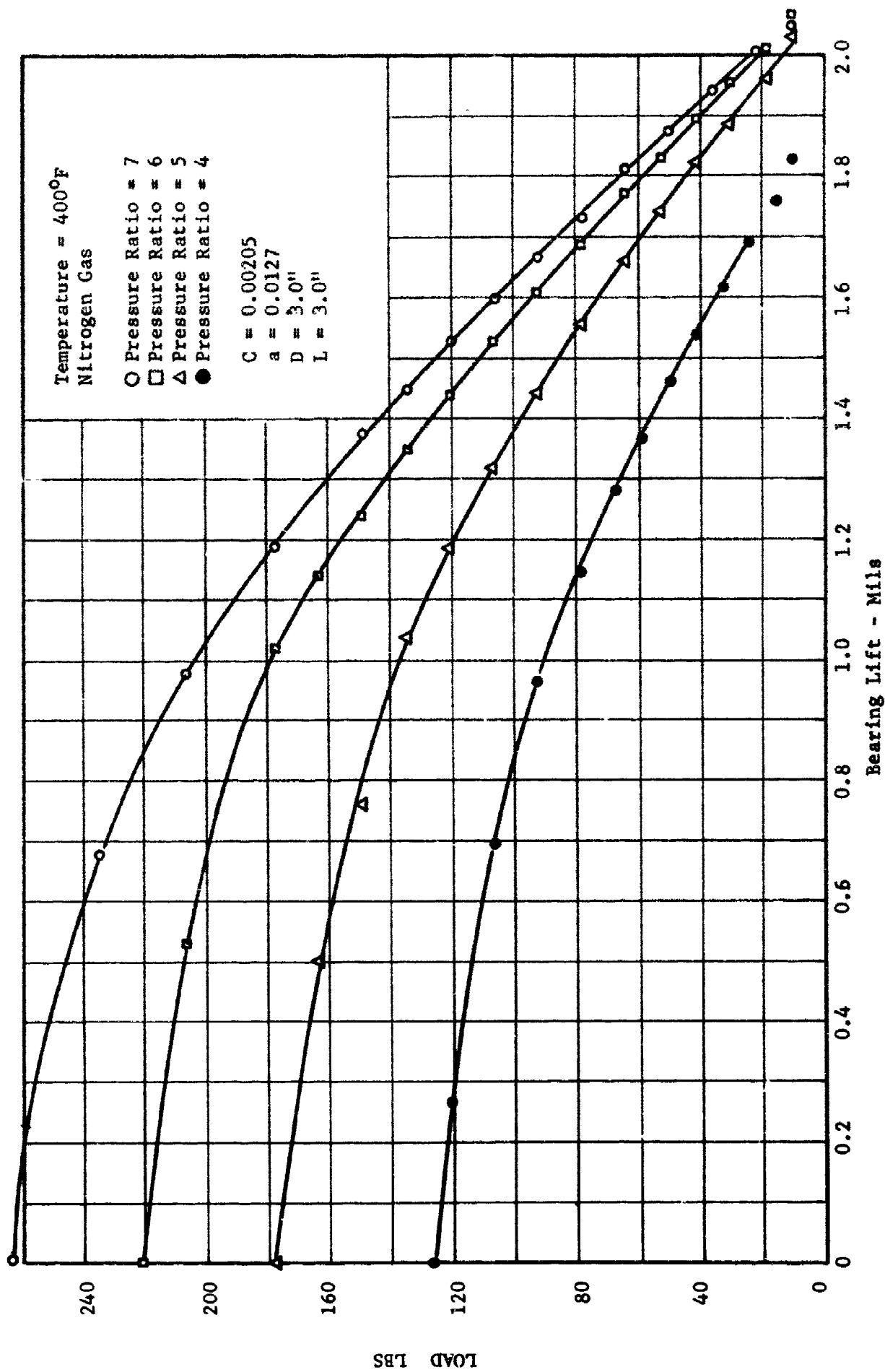


Fig. 4 Load-Deflection Curves, Nitrogen Lubricant, 400°F

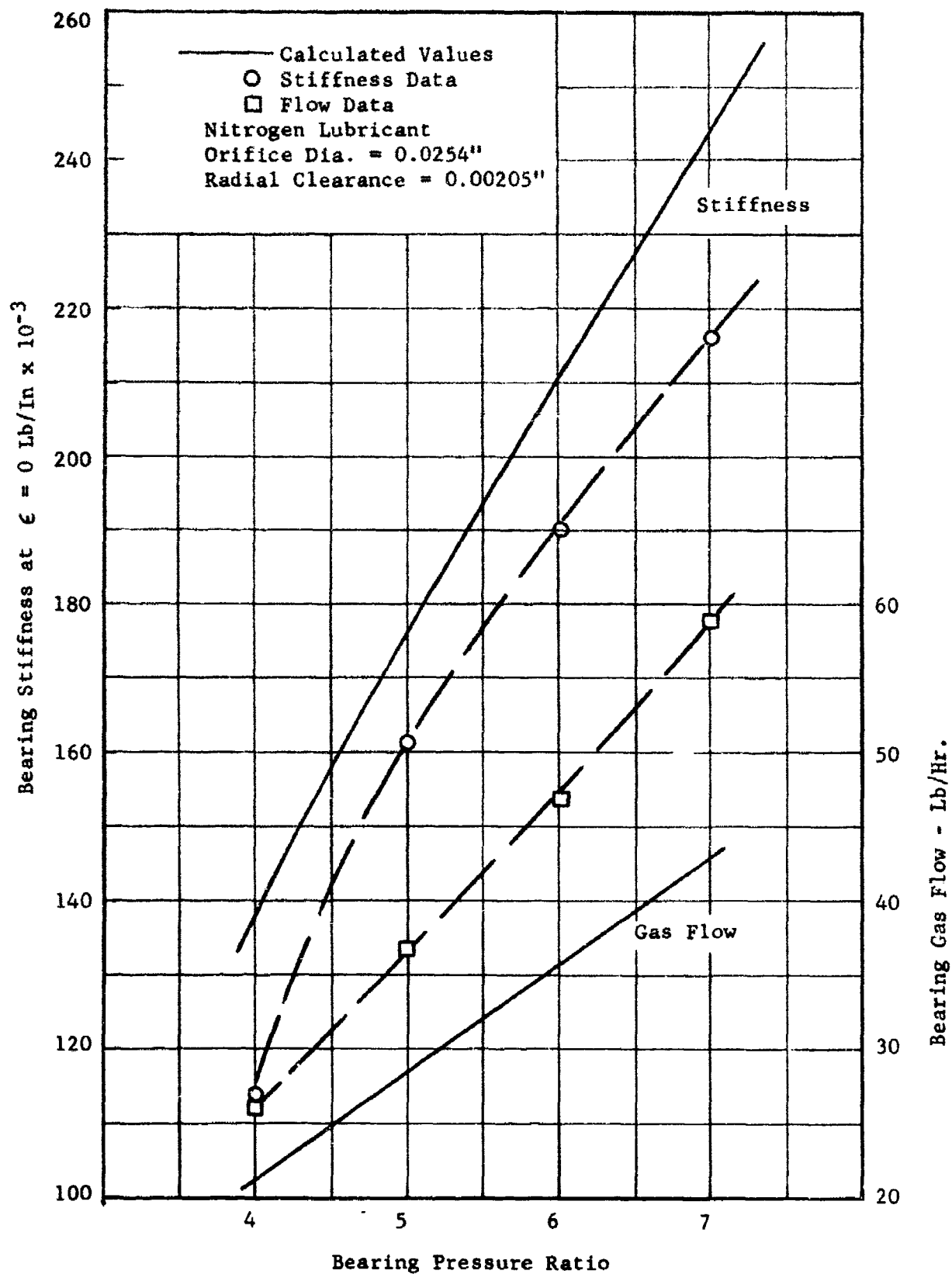


Fig. 5 Stiffness and Flow, Nitrogen Lubricant 400°F

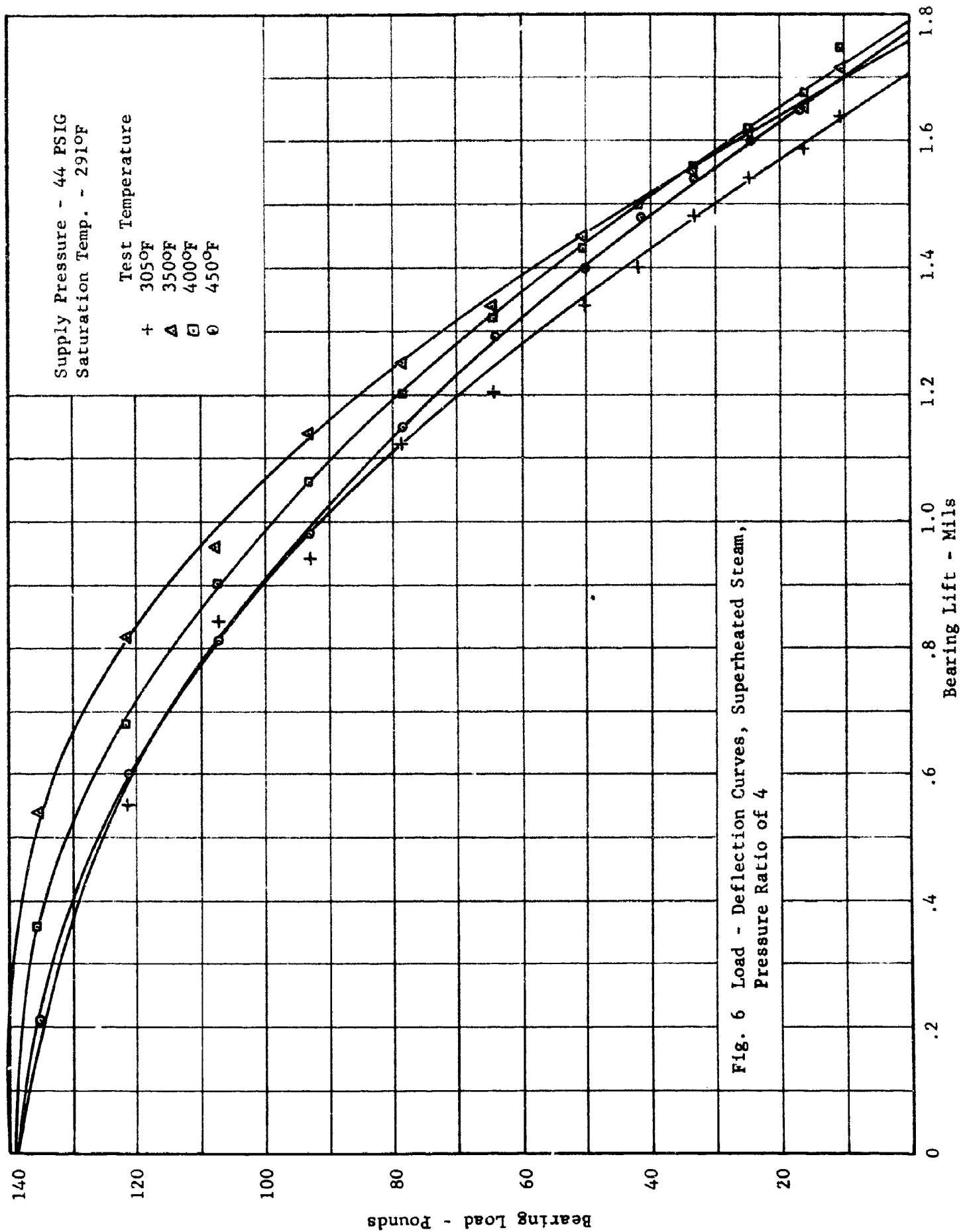


Fig. 6 Load - Deflection Curves, Superheated Steam,
Pressure Ratio of 4

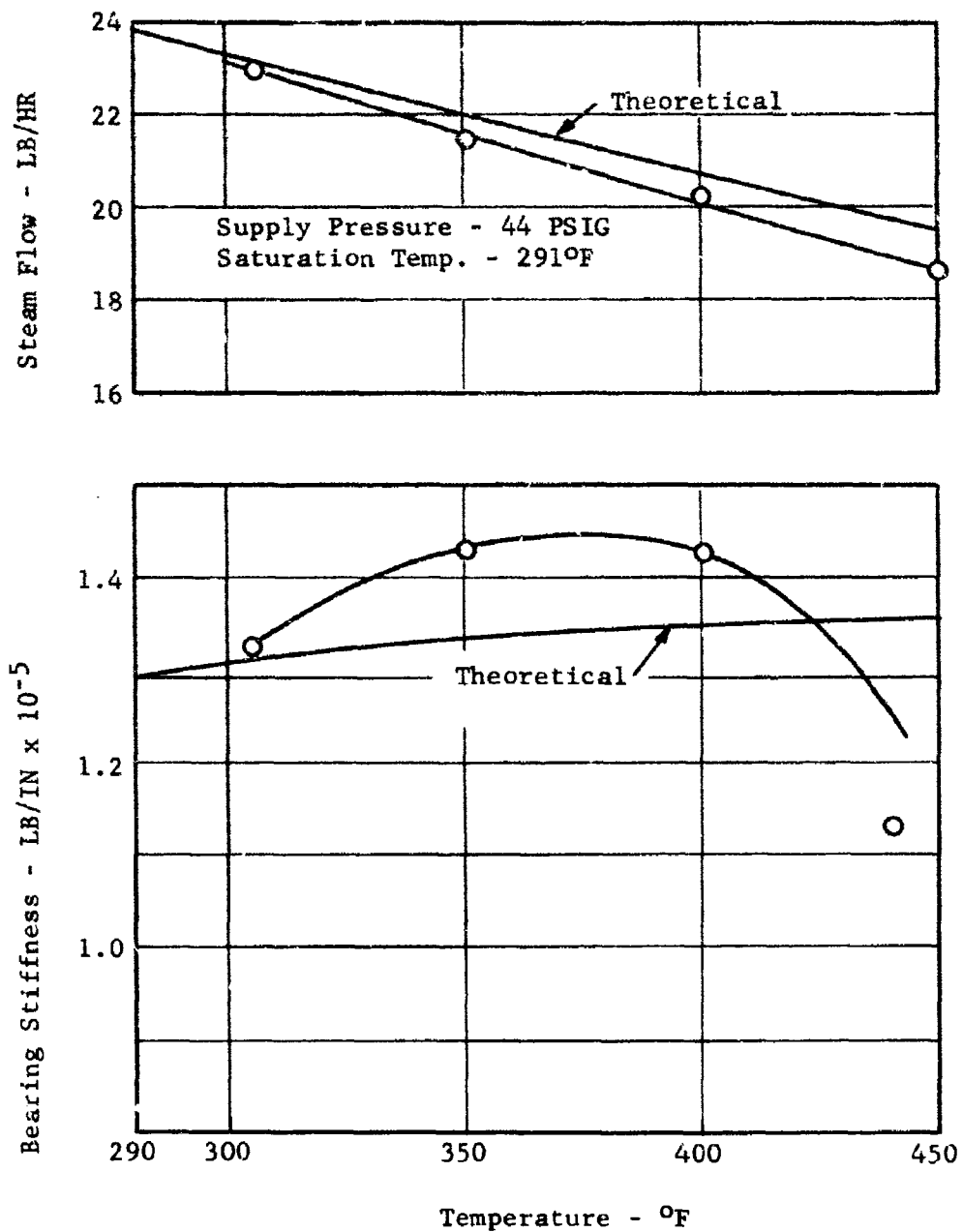


Fig. 7 Stiffness and Flow, Superheated Steam, Pressure Ratio of 4

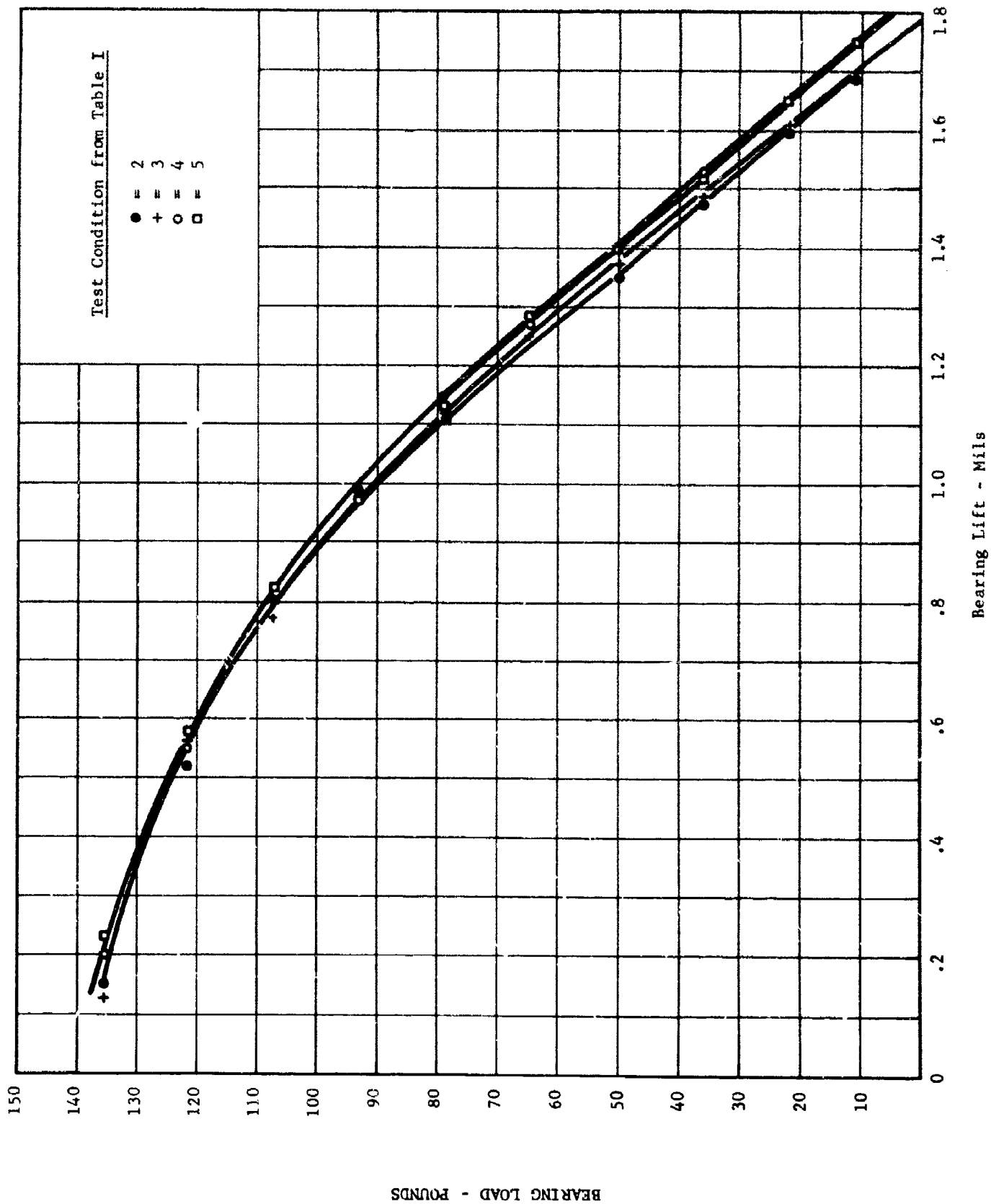
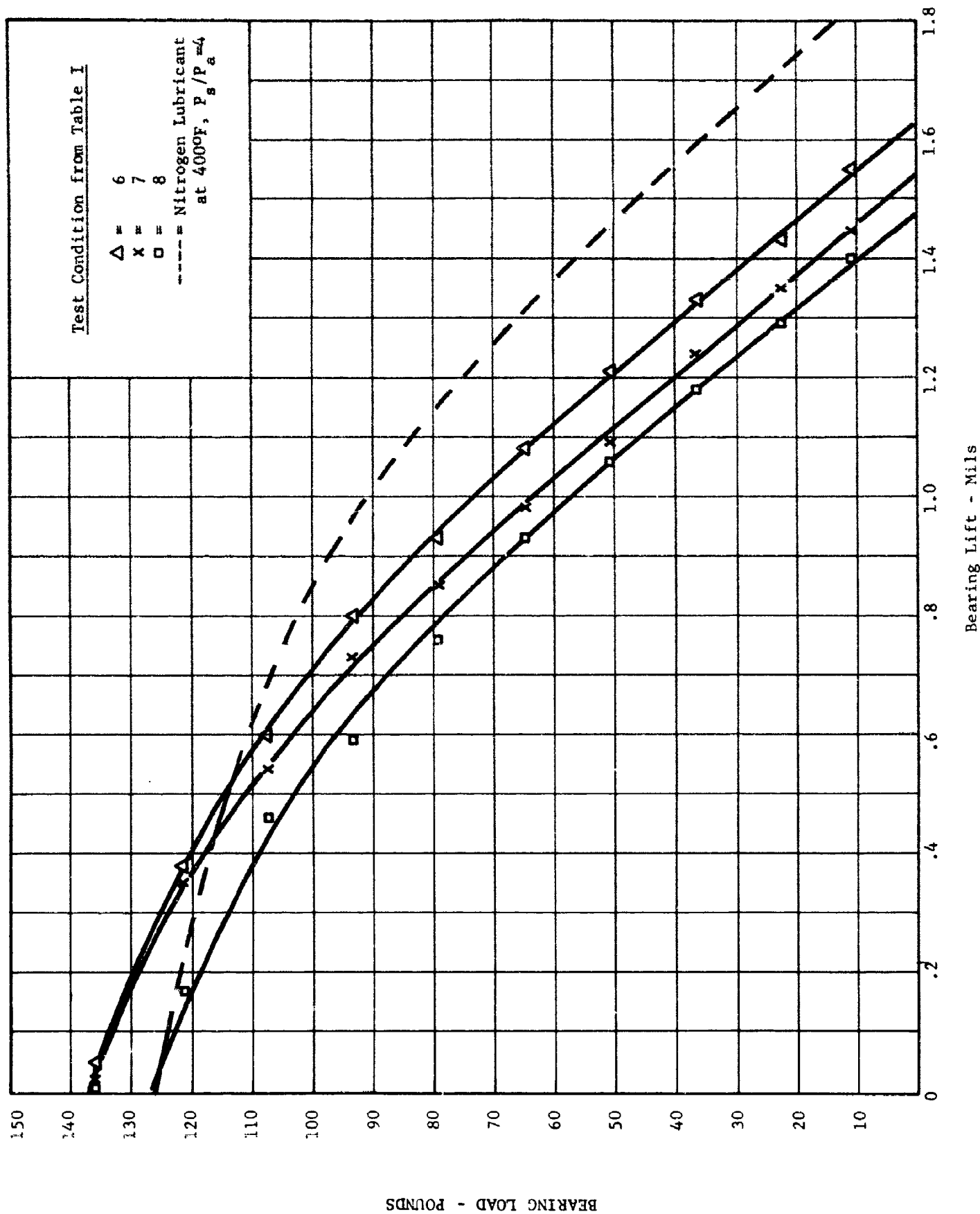
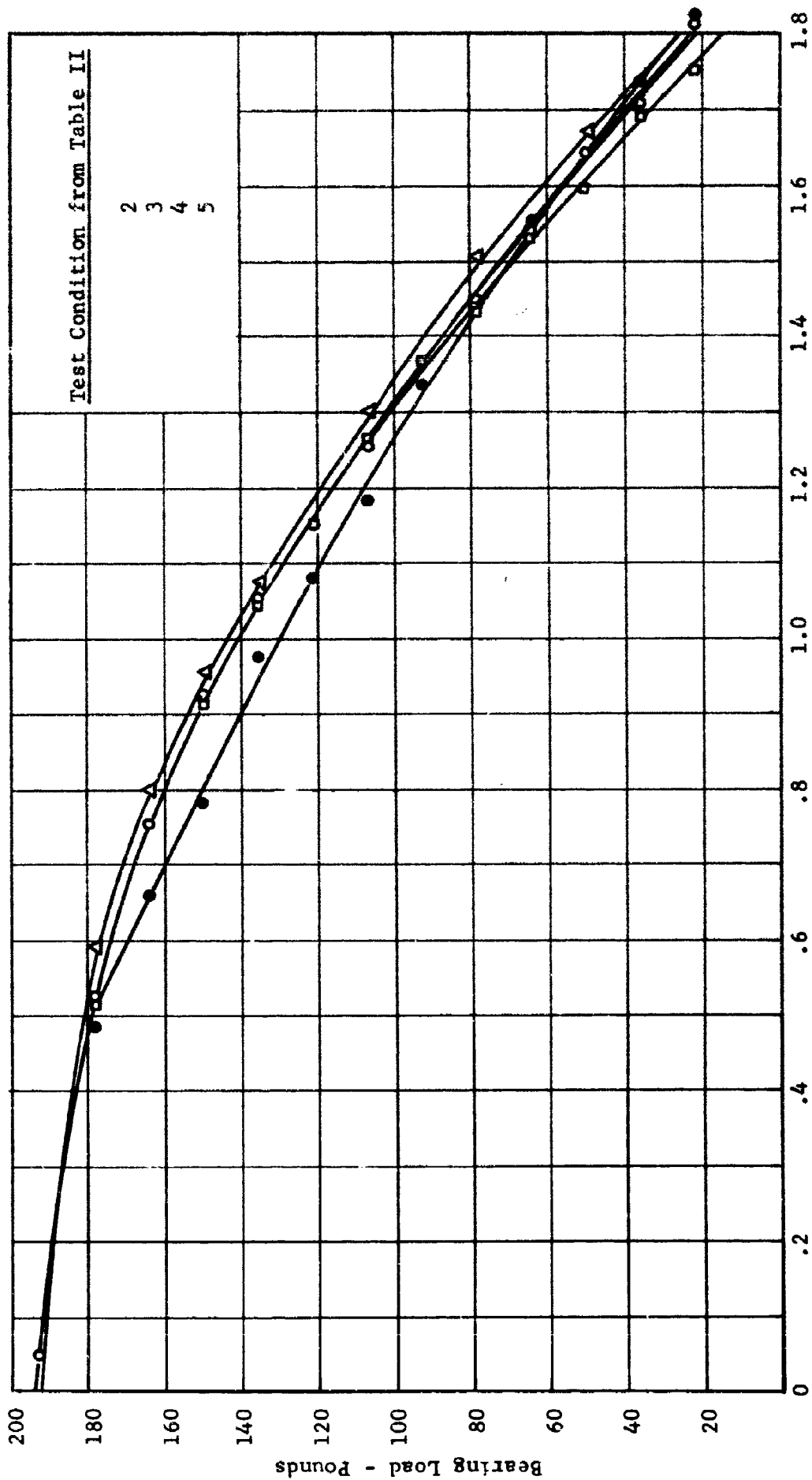


Fig. 8 Load - Deflection Curves, Steam, Near Saturation Temperature, $P_s/P_a = 4$





Bearing Lift - Mils

Fig. 10 Load - Deflection Curves, Steam, Near Saturation Temperature, $P_g/P_a = 5$

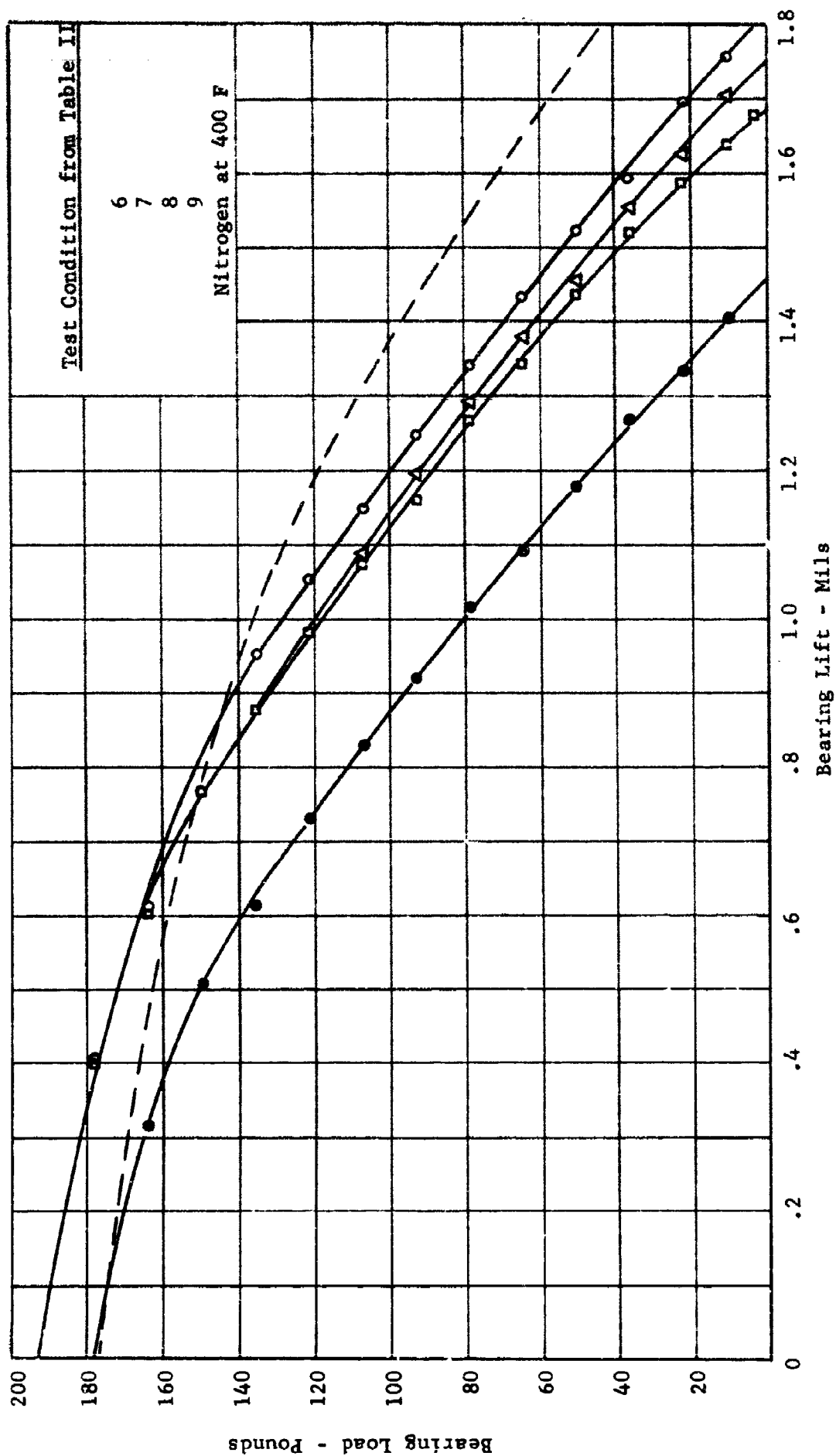
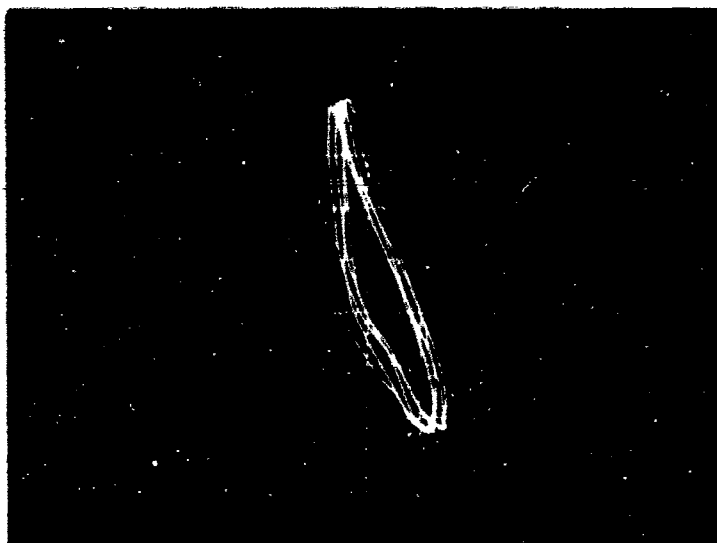


Fig. 11 Load - Deflection Curves, Steam, Near Saturation Temperature, $P_s/P_a = 5$, Cont.



Bearing Supply manifold pressure \approx 35 psig
Bearing temperature - 280°F
Shaft temperature - 265°F
Bearing Load \approx 20 lbs.
Inlet Steam Wet

Fig. 12 Instability "Orbit" - Wet Steam with Metal Temperature Below T_s

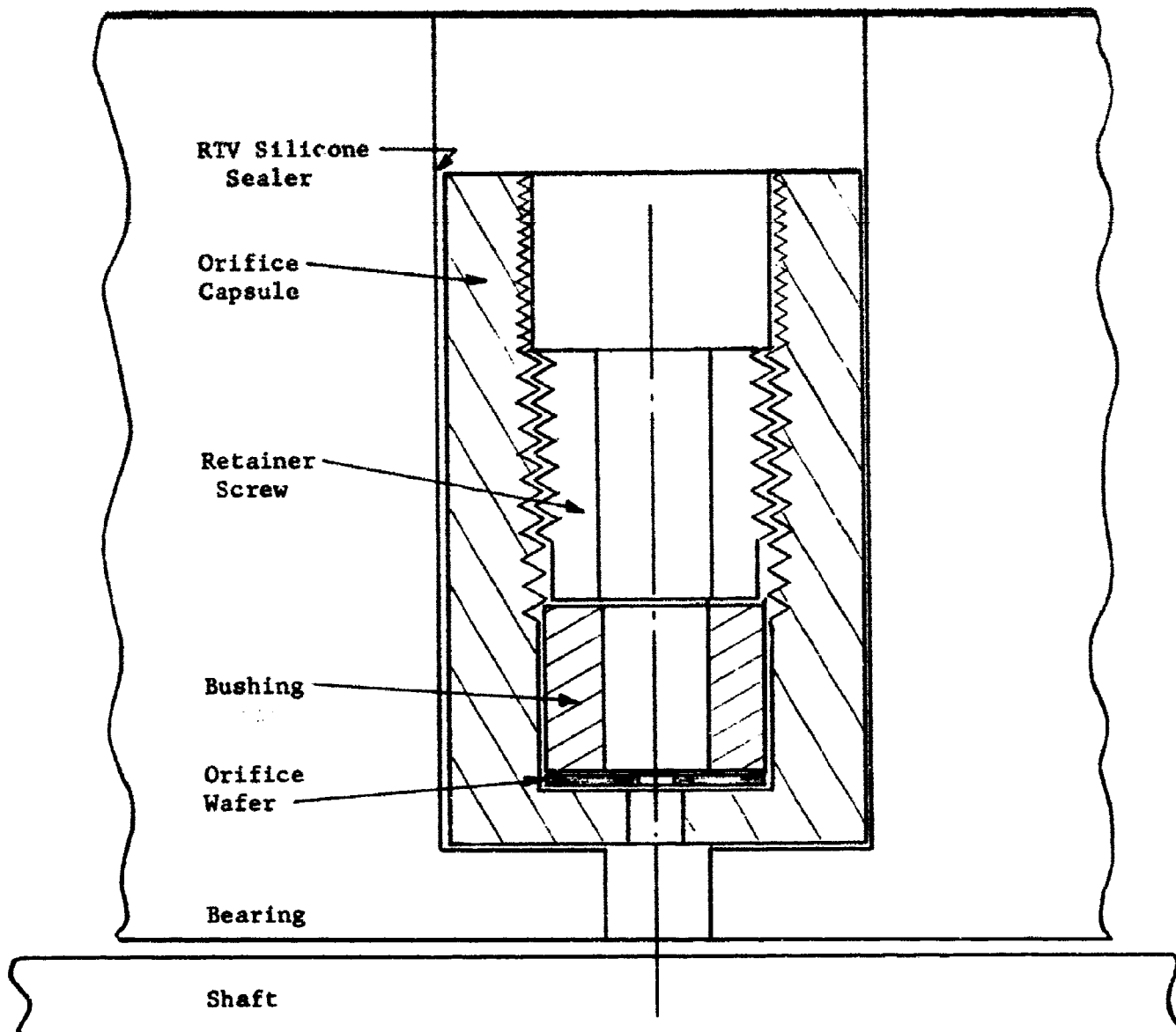


Fig. 13 Bearing Restrictor Orifice Installation

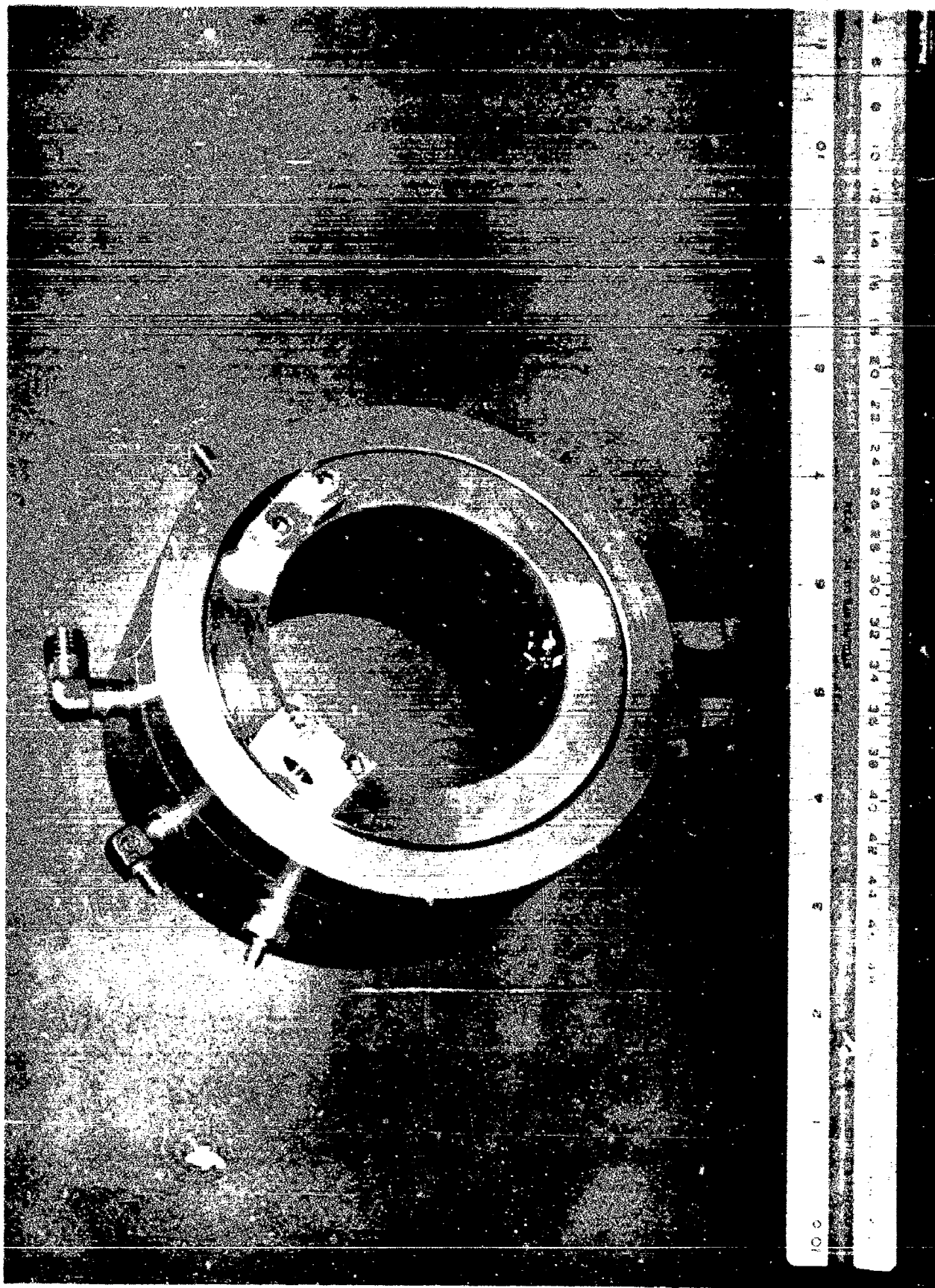
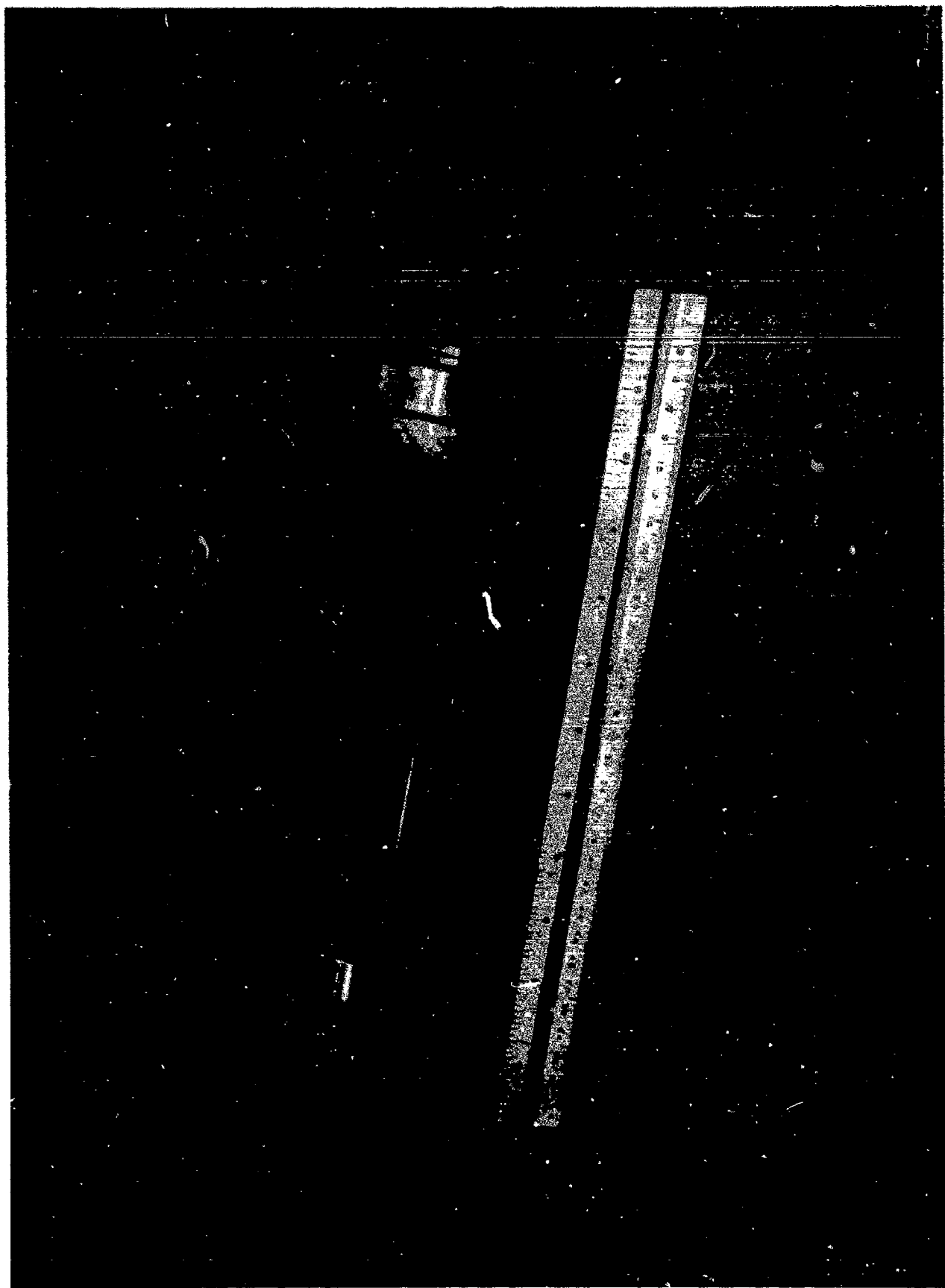
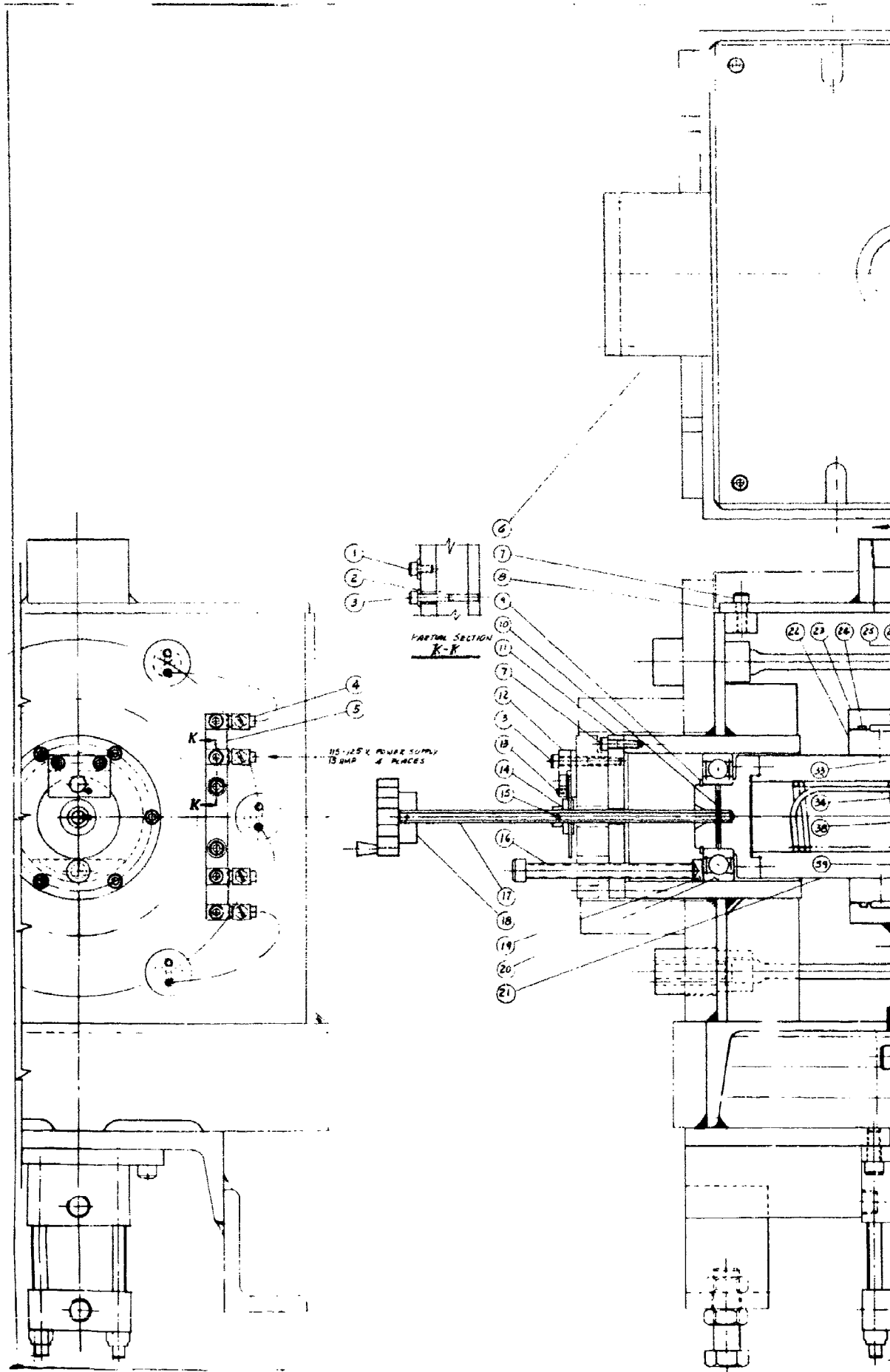


Fig. 14 Test Bearing Assembly



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Fig. 16 Assembly Drawing of Test App.

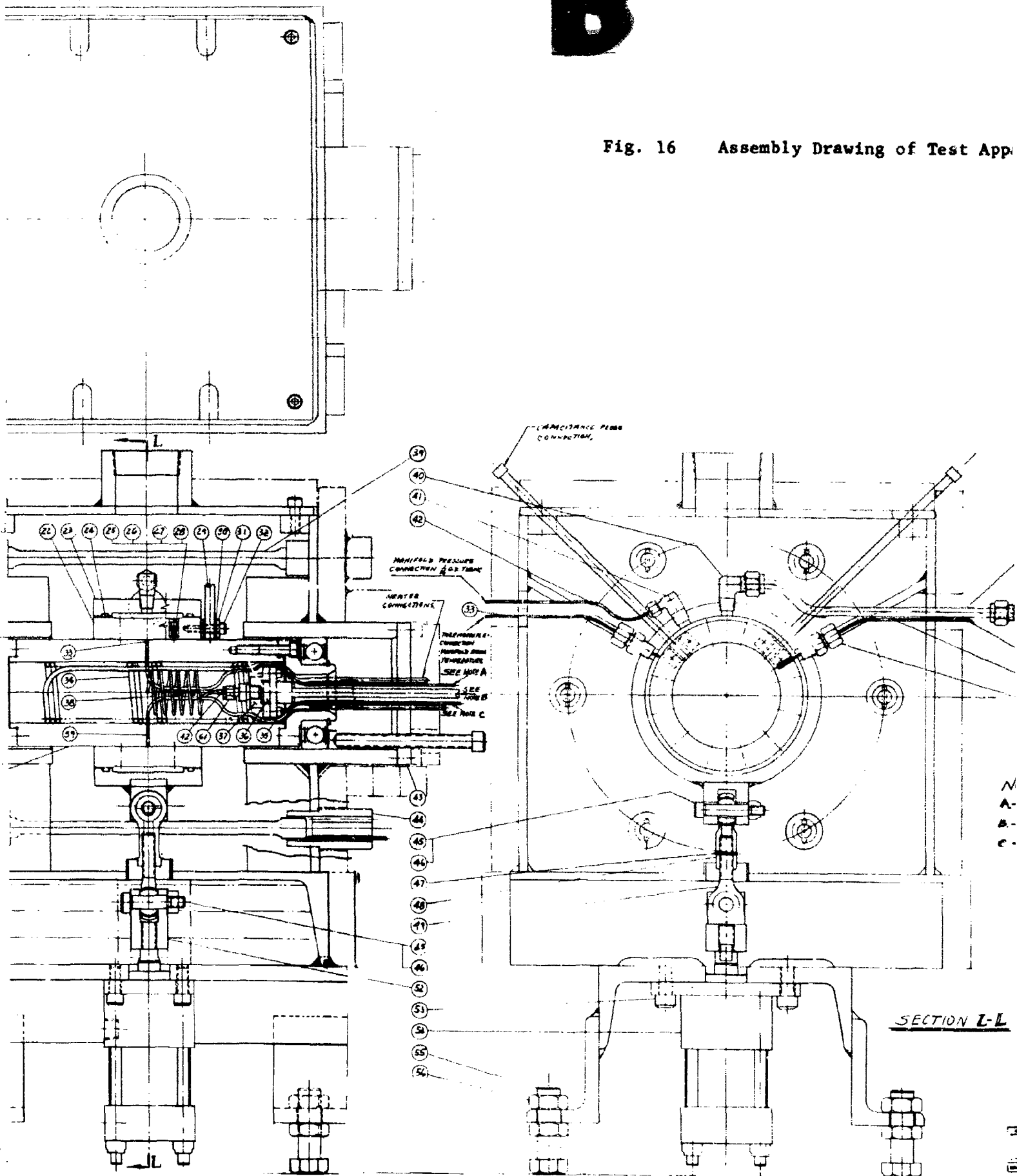
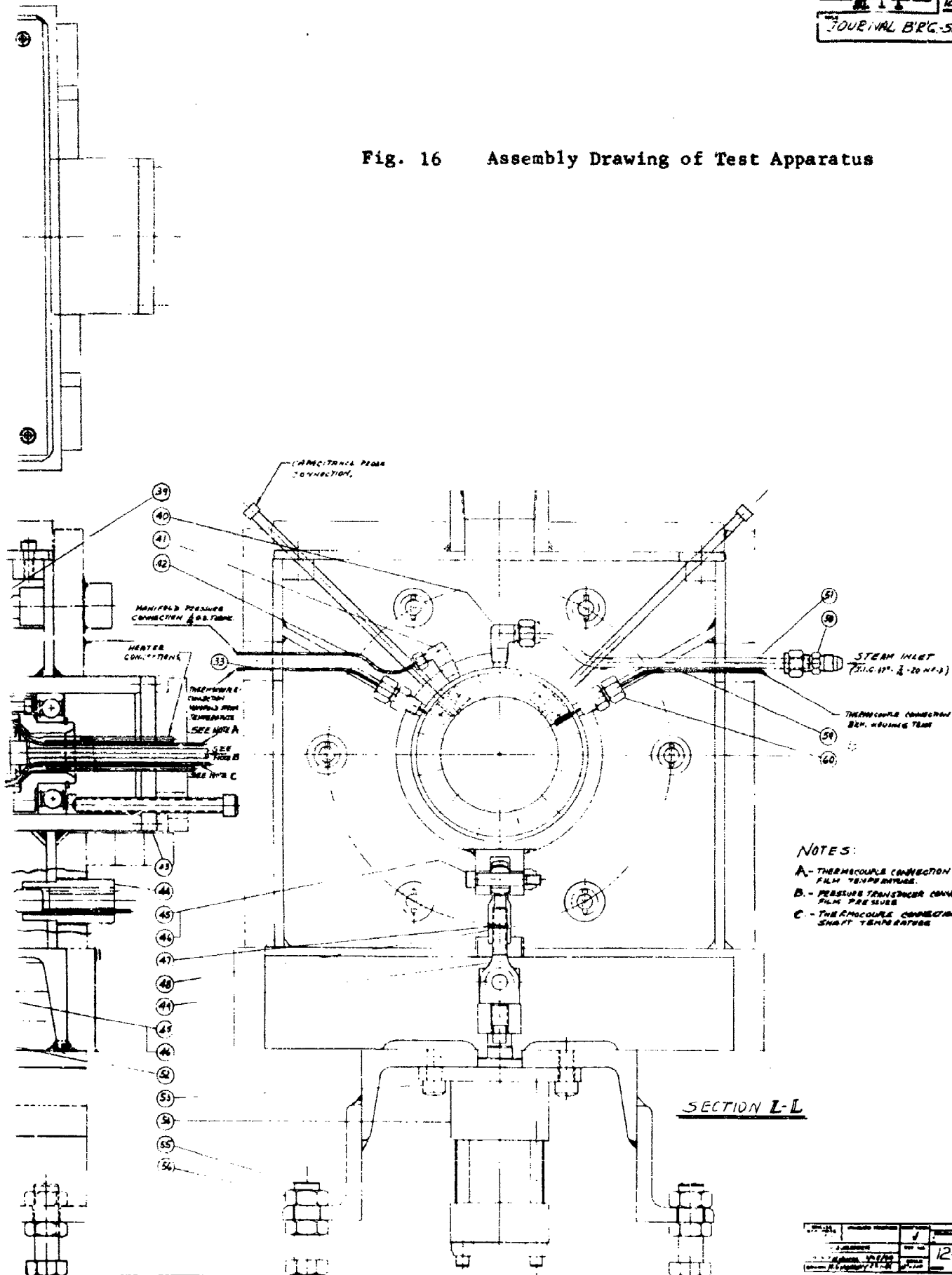


Fig. 16 Assembly Drawing of Test Apparatus



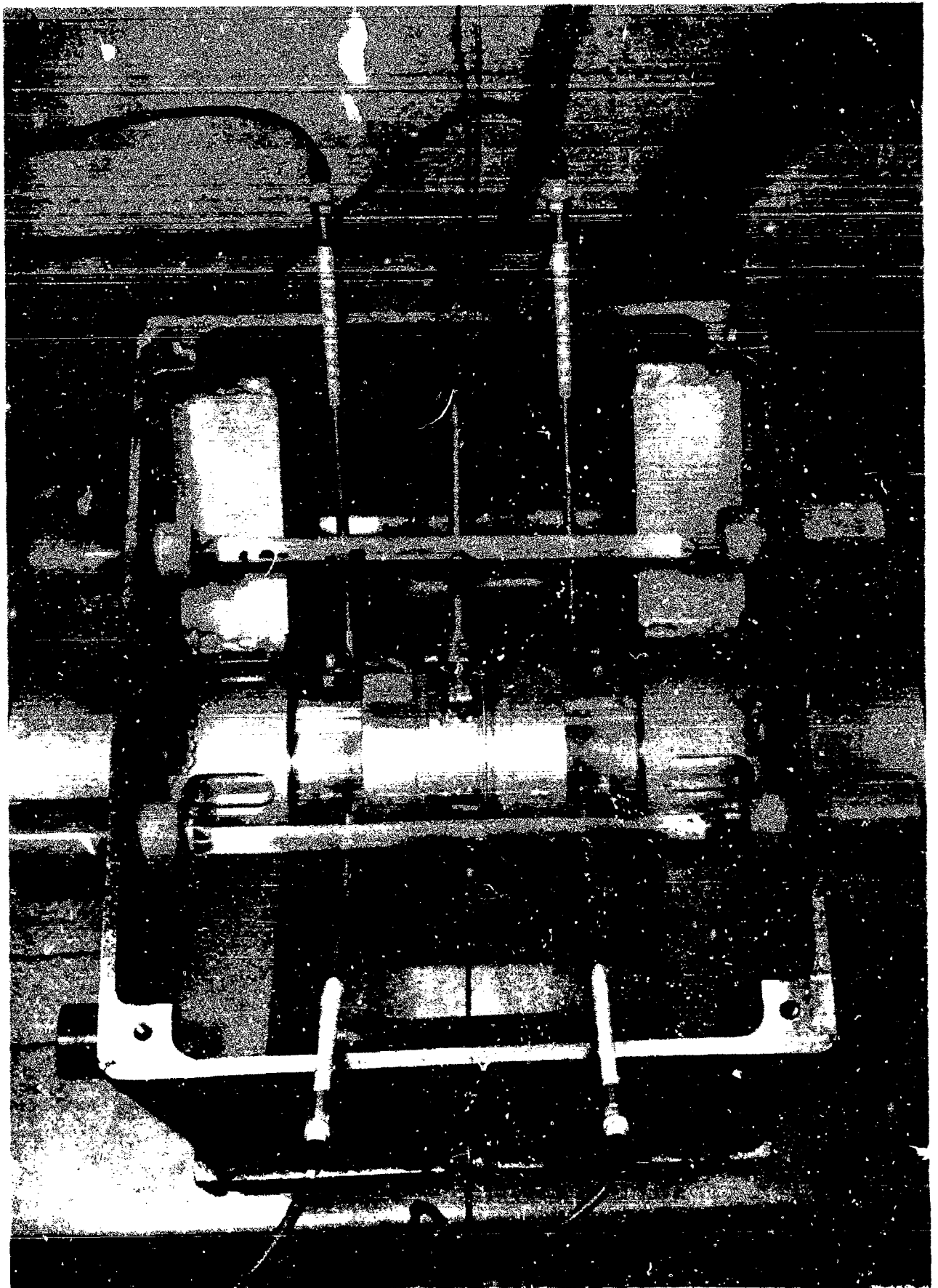


Fig. 17 Top View of Steam Journal Bearing Test Rig

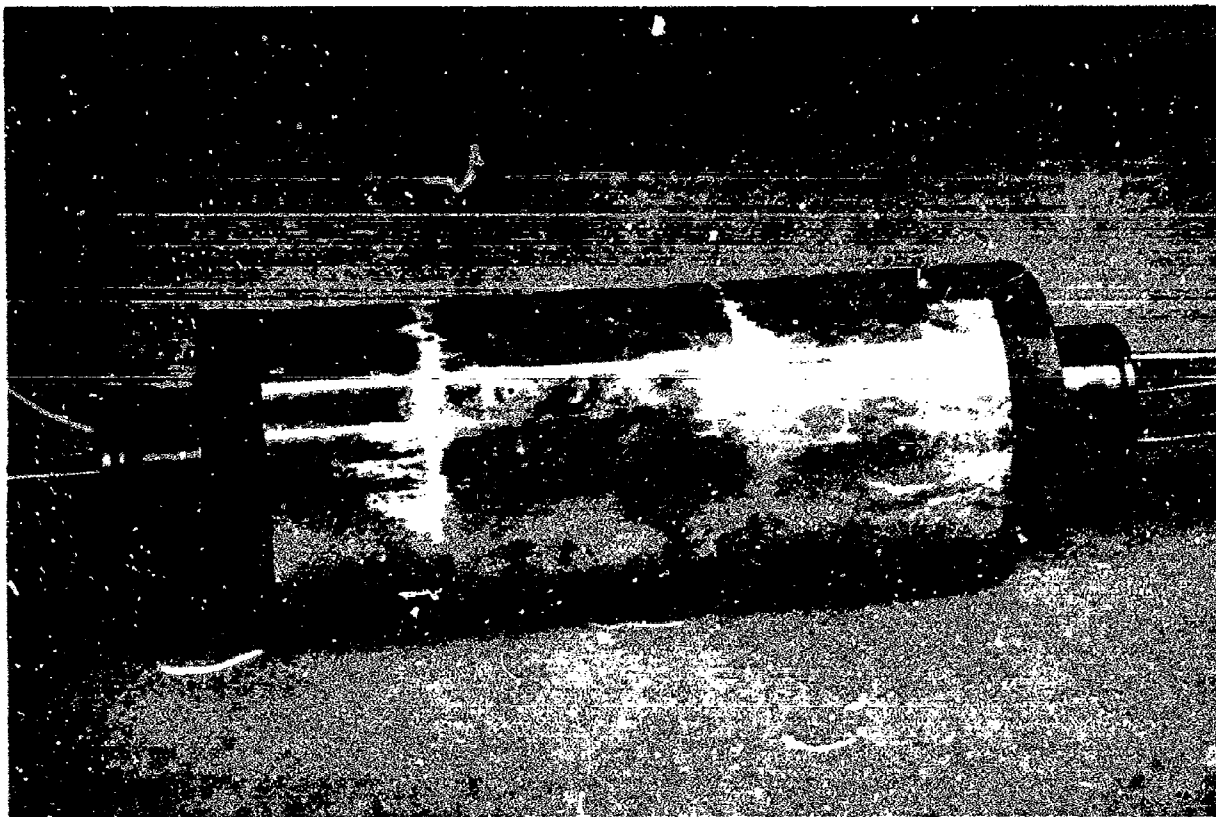


Fig. 18 Test Shaft with Mineral Deposits After Testing with Wet Steam

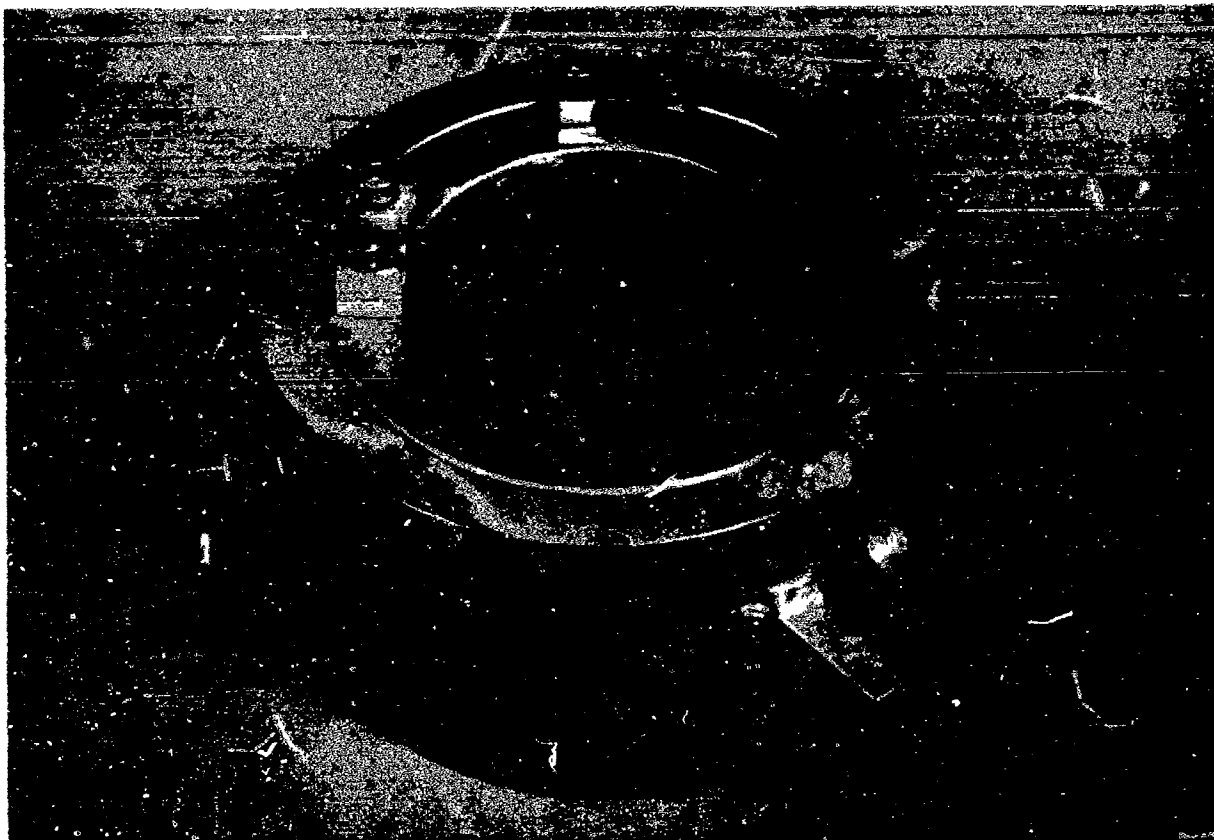


Fig. 19 Condition of Bearing After Testing With Wet Steam

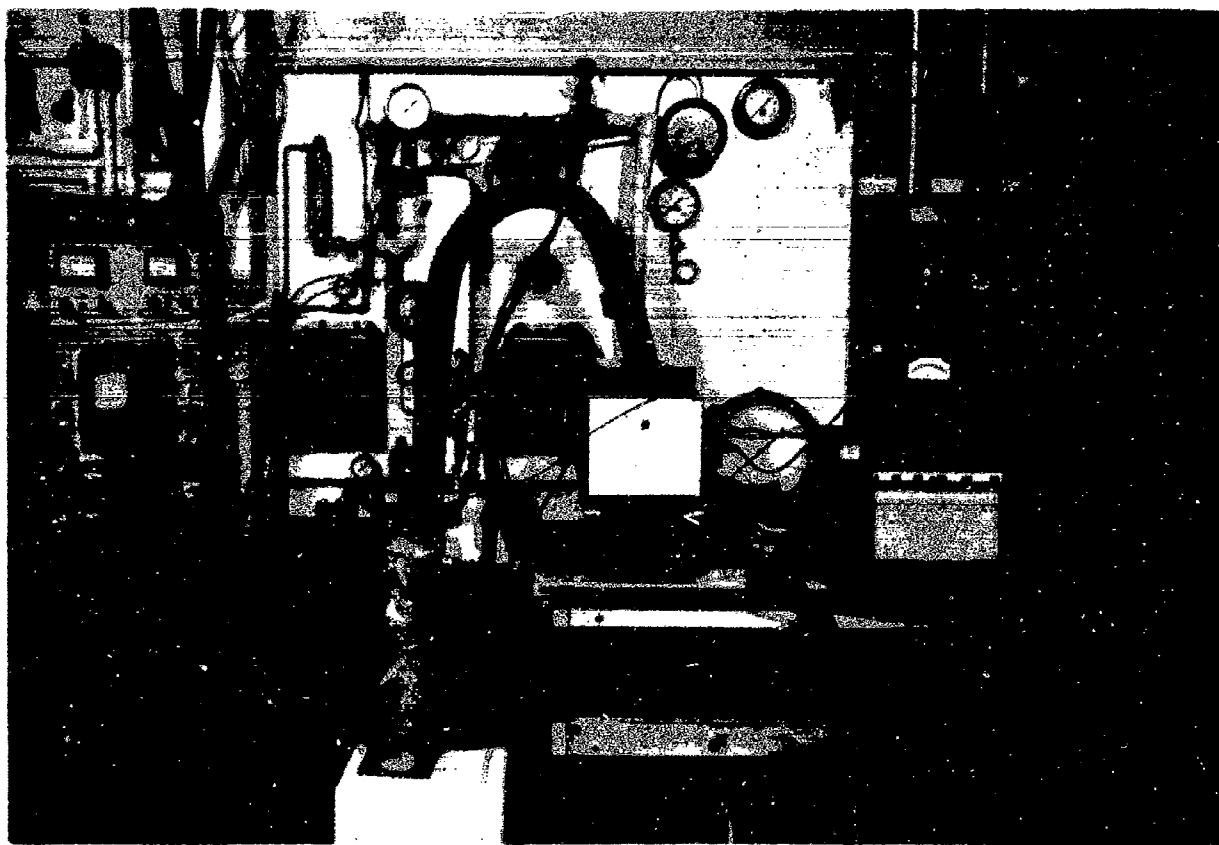
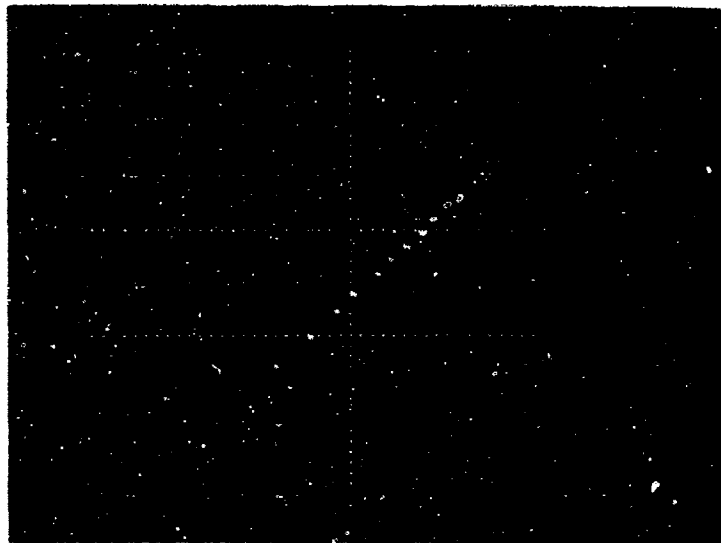


Fig. 20 View of Entire Test Facility



Pressure Ratio = 5
Supply Pressure = 59 psig
Inlet Steam Temperature = 313°F
Bearing Temperature = 305°F
Shaft Temperature = 306°F

Fig. 21 Typical Load Deflection Data

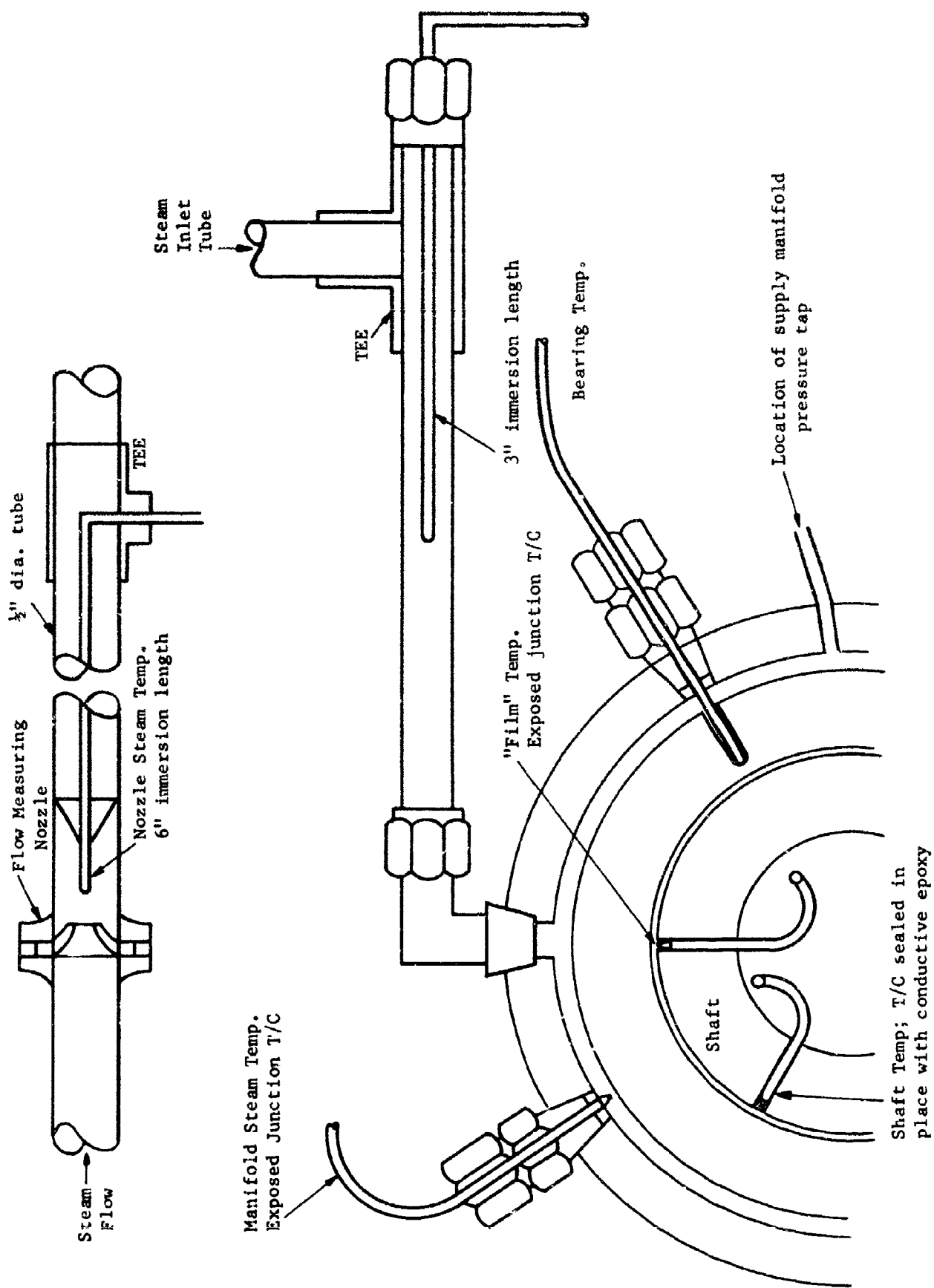


Fig. 22 Thermocouple Installations